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SOME NOTES ON GAS ENGINES.

BY PROF. H. DIEDERICHS, CORNELL UNIVERSITY.

[From *The Sibley Journal of Engineering.*]

If there is one movement of extreme importance in the engineering field of today it is the increased interest manifested in the gas engine as a source of motive power. Opinions as to the probable extent of this movement are various. Some will grudgingly concede that the gas engine is quite well enough developed to compete with the reciprocating steam engine in some fields, others, among them Diesel, claim with great assurance that the death knell of the steam engine has been sounded, and that it is but a matter of a few years when the steam engine will be a thing of the past. The impartial observer can not well agree with either the one or the other of these views. It is unquestionably a fact that the internal-combustion engine can show an immense improvement in the last ten years, and not the least among these the steps in advance, is the fact that gas engines can now be built for successfully operating alternating-current machinery in parallel, but to conclude from

this showing as a whole that the steam engine is absolutely doomed would seem to be going too far. It can, of course, not be denied that the internal-combustion motor will find much wider application than it now has, but it is quite safe to conclude that the conditions and exigencies of every case will decide between the gas engine and the steam engine, much in the same way as we decide today between high and low-speed, single or multiple-cylinder, condensing or non-condensing engines.

Practically every important industrial nation is concerned in this advance. It appears that we in America have been somewhat behind hand, and have allowed German and English manufacturers and designers to show us the way in some branches of this development, but, unless the signs fail, this state of things will not continue for any considerable length of time.

As an evidence of this movement, and in a measure of its importance, we find a large amount of scientific and practical discussion appearing in scientific journals and papers. In many cases these articles are of great importance and wide bearing, and this is especially true when they come from the pens of such men as Humphrey and Burstall in England, Witz and Hubert in France and Belgium, and Meyer, Schöttler, Schröter, Güldner, Diesel and others in Germany. It is not an easy matter for any engineer to keep himself posted and abreast of the times on engineering matters concerning his own country. Much more difficult is the task when he attempts to do the same thing in a foreign language with which he may be little familiar. Nevertheless the importance of this subject of gas-engine development and the fact that some other nations at the time of this writing have admittedly a slight lead in this field, would justify some pains in this direction.

It is the purpose of the following paragraphs to report upon and discuss briefly some of the interesting tests and treatises that have appeared in German literature within the last two years, together with some data which have come under the writer's own observations. After what has been said above it is hardly necessary to further justify the presentation of this

matter in this shape, and while no doubt much of the following is familiar to some, it is hoped that some service may be done.

I. GAS ENGINE CYCLES, THEIR THEORETICAL THERMAL EFFICIENCIES AND PRACTICAL LIMITATIONS.

On examination it will be found that all practical internal-combustion motors work with one or the other of two cycles. The first class employs the Otto cycle, the second a cycle the type of which is the Diesel cycle of today. In our discussion we will call this the Diesel Cycle. This classification holds good whether the engines employ the 4-cycle or 2-cycle principle. Since the Diesel of today is a 4-cycle engine, we will further limit this discussion to the 4-cycle engine, it being understood, however, that the thermodynamic expressions derived hold just as well for the 2-cycle type.

1. THE OTTO CYCLE.

In this cycle heat is added along the line 1—2 at constant volume. Adiabatic expansion takes place along 2—3. The heat is rejected along line 3—4 at constant volume. The remainder of the charge is displaced along 4—5, while a new charge is drawn in along 5—4. This is adiabatically compressed along 4—1, completing the cycle. In the theoretical cycle, the work along 4—5 equals that along 5—4, and since neither of these changes have any connection with the heat interchanges in the cycle proper, the cycle to be examined consists of the lines 4–1–2–3–4.

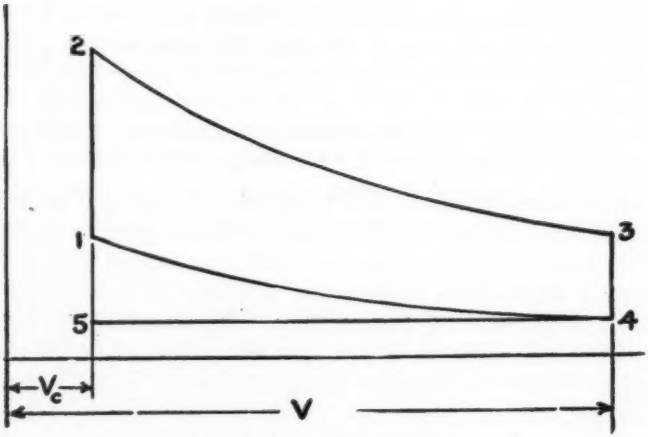
Let H_s be the heat supplied to the cycle.

Let H_r be the heat rejected.

Let $p_1, p_2, \cdots, v_1, v_2, \cdots T_1, T_2, \cdots$, be the absolute pressures, volumes and absolute temperatures, respectively, at the points 1, 2, $\cdots$.

Finally, if K_p and K_v be the specific heats of the charge at constant pressure and constant volume, respectively,

$$\text{let } x = \frac{K_p}{K_v}.$$



The cyclic, or the possible thermal efficiency of any cycle, is

$$E = \frac{H_s - H_r}{H_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Substituting in (1), the values of H_s and H_r determined from well-known thermodynamic conditions for this cycle, we shall have

$$E = \frac{(T_2 - T_1) - (T_3 - T_4)}{T_2 - T_1}$$

$$= \frac{T_2\left(1 - \frac{T_3}{T_2}\right) - T_1\left(1 - \frac{T_4}{T_1}\right)}{T_2 - T_1}.$$

But $\frac{T_2}{T_1} = \frac{T_3}{T_4}$ from which $\frac{T_4}{T_1} = \frac{T_3}{T_2}$

$$\therefore E = \frac{(T_2 - T_1)\left(1 - \frac{T_4}{T_1}\right)}{T_2 - T_1} = 1 - \frac{T_4}{T_1} = 1 - \frac{T_3}{T_2}.$$

This may also be written

$$E = 1 - \left(\frac{V_c}{V}\right)^{x-1}$$

If we let $\frac{V}{V_c} = \frac{\text{total volume at end of stroke}}{\text{clearance volume}} = r$ we shall finally have

$$E = 1 - \left(\frac{1}{r}\right)^{x-1} = 1 - \frac{1}{r^{x-1}}.$$

This expression shows that the efficiency of this cycle depends upon the values of r and x. It shows that as r is increased, *i. e.*, as the clearance volume is decreased, the efficiency will increase. Further, as x increases the efficiency will also increase. Now x depends upon the composition of the mixture, for any given fuel, gas or oil, x being lower the richer the charge. So that this expression finally shows that the leaner fuel mixtures may be expected to give the better results.

This fact is well illustrated in the following figures.*

Ill. Gas	*Mixture of Air*	$r=$	$x=$	$E=$
1	6	5	1.354	.435
1	13.5	5	1.383	.461

This shows a difference of 6 per cent. in favor of the poorer gas mixture.

Lastly, since the efficiency depends only upon r and x, it is independent of the maximum temperature in the cycle, *i. e.*, it is independent of the load on the engine as long as r and x remain the same.

II. THE DIESEL CYCLE OF TODAY.

In this cycle we have heat added along 1—2 at constant pressure, and adiabatic expansion from 2—3. The heat is rejected at constant volume along 3—4. The charge is displaced along 4—5, and a new charge drawn in along 5—4. The final step is adiabatic compression along 4—1.

The best treatment of this case would seem to be the following: †

As before $$E = \frac{H_s - H_r}{H_s},$$

$$H_s = K_p (T_2 = T_1),$$

$$H_r = K_v (T_3 = T_4).$$

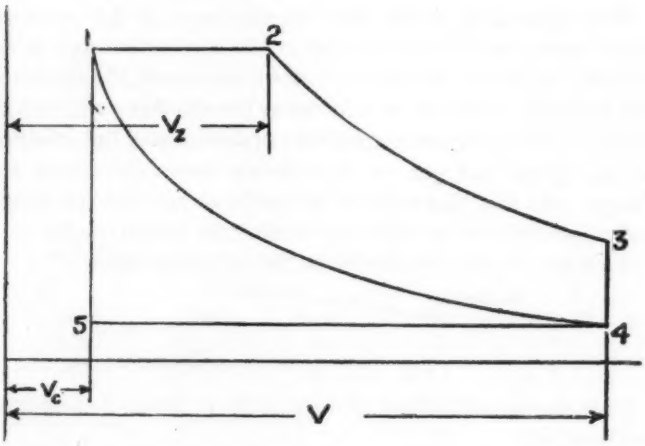
Expressing T_2 in terms of T_1, and T_3 in terms of T_4, we have

$$T_2 = T_1 \frac{V_z}{V_c}$$

and $$T_3 = T_4 \left(\frac{V_z}{V_c}\right)^x$$

*Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 156.

†Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 160.



If we put $\frac{V_2}{V_c} = \delta$, we will have

$$T_2 = T_1\delta$$

and $$T_3 = T_4\delta^x$$

Substituting the above in the expression for E, we have

$$E = 1 - \frac{K_v}{K_p}\left(\frac{T_4\delta^x - T_4}{T_1\delta - T_1}\right)$$

$$= 1 - \frac{K_v}{K_p}\frac{T_4}{T_1}\left(\frac{\delta^x - 1}{\delta - 1}\right),$$

and since $\frac{K_p}{K_v} = x$, and $\frac{T_4}{T_1} = \left(\frac{V_c}{V}\right)^{x-1} = \frac{1}{r^{x-1}}$

we have finally

$$E = 1 - \frac{1}{r^{x-1}}\frac{1}{x}\left(\frac{\delta^x - 1}{\delta - 1}\right).$$

This equation differs from that obtained for the Otto cycle by the factor $\frac{1}{x}\left(\frac{\delta^x - 1}{\delta - 1}\right)$ in the last term.

A brief inspection will show that with values of δ and x greater than 1, which always holds true in this case, this factor as a whole is greater than one. That means that on purely thermodynamic grounds, with the same degree of com-

pression, the Otto cycle will show a greater cyclic efficiency than the Diesel cycle. The cyclic efficiency of the latter depends not only upon r and x, but also upon δ, and it decreases as δ increases, *i. e.*, as the load on the engine increases. This fact may serve to explain why Diesel engines on numerous tests have shown an increase in the thermal efficiency as the load decreased from the normal.

It may be remarked that the formula is not rigidly correct for this cycle, as we proceeded on the assumption that the weight of charge remains the same throughout, which is not the case. But the increase in the weight of charge for oil fuel is very slight, while it might be considerable for a gas fuel. The only successful machine employing this cycle, the Diesel, is an oil engine, so that this is the only case that need be taken into account. The error made in applying the formula to this case is therefore small, and would tend to somewhat further decrease the efficiency.

For a pressure of 475 pounds absolute at end of compression, and values of x and r as assumed in the table, the following figures show how the efficiency depends upon a variation in δ, *i. e.*, in the load on the engine:

$\delta =$	1.5	2.0	2.25	2.50	2.75	3.00
$x = 1.30$, $r = 15$.....	$E = .532$	.503	.490	.478	.468	.459
$x = 1.40$, $r = 10$.....	$E = .567$	.535	.519	.507	.494	.483

The cyclic efficiency of the Otto cycle, for the same conditions of x and r, would have been constant for all loads at .556 for the first case, and .602 for the second.

The Otto cycle is therefore on theoretical grounds the one to use, provided it can work with the same compression ratio as the Diesel. Unfortunately practical difficulties step in which make some of the theoretical conditions assumed impossible to realize in practice. The most serious difficulty is the fact that pre-ignition of the charge may occur during compression, if the compression is carried too high. The point at which pre-ignition is apt to occur depends upon the composition of the charge, and is further modified by the ac-

tion of the cylinder walls in any particular case. It may be said in general that the higher the contents of the charge in hydrogen the easier will pre-ignition occur, *i. e.*, the lower must the compression be kept for satisfactory working. A glance at the efficiency formula for this cycle will show that this difficulty at once sets a limit to the attainable efficiency. The Diesel cycle is not open to this objection because here air alone is compressed, fuel and air being kept separate until combustion is desired. In fact, the satisfactory working depends here upon a sufficiently high air temperature, and the limit set is only the fact that high pressures cause trouble in mechanical operation.

The conditions being thus outlined, it becomes interesting to inquire to what extent the conclusions drawn on theoretical grounds are thereby modified. To secure a basis of comparison, it may be assumed that the maximum pressures realized in both cycles are the same. Although the maximum pressure maintained in the Otto but momentarily, the machine parts have to be designed for this pressure, and hence the maximum pressure really sets the upper limit for either cycle. Further, under nominal load, the Diesel cycle works with about 10 per cent. filling of the cylinder, and from this the heat supplied each cycle may be computed. The temperature at beginning of compression is taken at 600 degrees absolute. The specific heat at constant volume of the mixture is assumed at $K_v = .19$, that for constant pressure at $K_p = .26$, making the ratio $\frac{K_p}{K_v} = 1.37$, which may be assumed an average case. It is assumed that the Otto cycle for the same maximum pressure receives the same amount of heat as the Diesel, and on this basis a range of efficiencies has been computed. These figures show what each type of cycle can do when each is working under the same maximum pressure and is furnished the same amount of heat, this amount of heat being assumed that necessary to generate the nominal power of the engine under the conditions chosen.

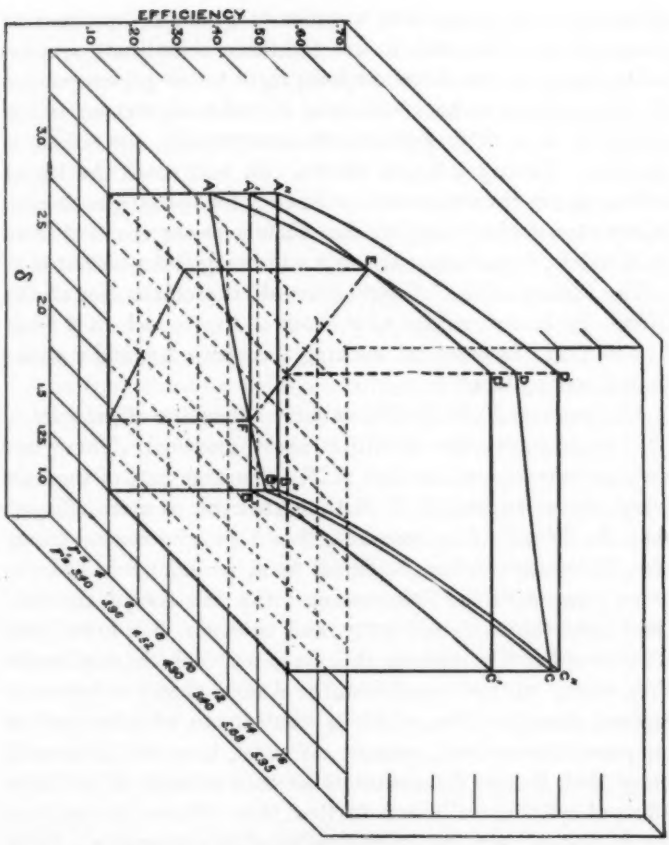
There being three variables present in the formula for the

Diesel cycle, it seems best, in order to give a comprehensive presentation of the field, to use the three co-ordinate system. In the figure shown, distances from right to left present values of δ, from front to back values of r, and from bottom to top values of E^*. The methods of constructing the solids is obvious. Three solids are shown. In each case the upper surface shows the variation in efficiency for the ranges chosen. In the case of the Otto cycle the efficiencies are constant from right to left, depending only on r and being independent of δ.

The surface $A\ B\ C\ D$ represents all the efficiencies of the Diesel cycle for values of r from 4 to 16, and of δ from 1.25 to 3.00. In practical working δ is about 2.5, and r varies from about 13 to 16.

The surface $A'\ B'\ C'\ D'$ similarly shows the efficiencies of Otto cycle under the condition above assumed. These two surfaces intersect in the line $E\ F$. For that part of the field lying in the surface $A\ E\ F$, the Otto cycle is more efficient than the Diesel; *i. e.*, assuming $\delta = 2.5$ as the normal condition, the value of r for the Diesel must be about 8 in order to be on a par with the Otto engine. For the rest of the field the Diesel shows a superiority, and since $r = 8$ is lower than what is ordinarily used for this cycle we reach the conclusion that, under normal condition, the Diesel shows a better efficiency than the Otto, which is contrary to what is obtained on purely theoretical ground. It must, however, be remembered that in this discussion no account is taken of probable effect of cylinder walls, and, further, that different assumptions for K_p and K_v will lead to somewhat different results. Güldner, for instance, on the basis of some computations comes to the conclusion that the two cycles when worked under the same maximum pressure show but a slight advantage in favor of the Diesel, so slight that a small increase in the filling of the cylinder above 10 per cent. reverses the results. To realize this the intersection $E\ F$ would have to be much nearer to the diagonal $D'\ B'$. The difference is quite likely

* r' represents the equivalent ratio of compression for the Otto cycle. As an example, if $r = 10$ for the Diesel cycle, the Otto cycle having the same maximum pressure, 352 pounds, and the same amount of heat furnished per cycle, 329 *B. T. U.*, will have a ratio of compression $r' = 4.90$.

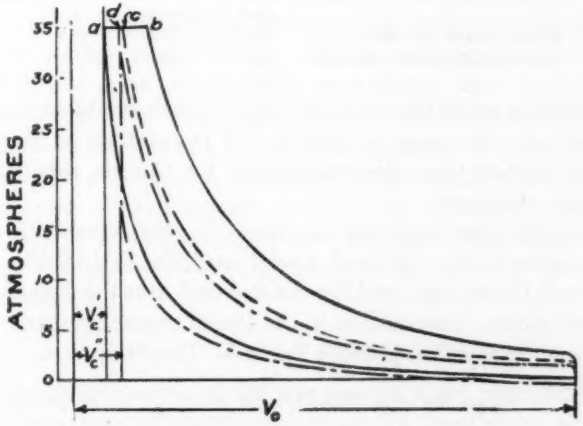


due to a difference in the assumptions. On the basis of his results Güldner is inclined to explain the superiority of the Diesel, as regards thermal efficiency on the indicated horse-power as realized in actual practice, not so much on the ground of superiority of cycle, but on the ground of complete vaporization of fuel and complete and perfect mixture of fuel and air.

To complete the picture, the surface $A''\ B''\ C''\ D''$ has been constructed, showing the efficiencies of the Otto cycle under

the same compression ratios as the Diesel. This is, of course, as explained, the ideal case. The efficiencies are throughout higher than those of the Diesel cycle.

Attempts have been made to overcome the difficulty of pre-ignition in the Otto cycle by making the composition of the charge such that pre-ignition is not likely to occur easily. The most successful of these is the device of Professor Donat Banki, who uses a mixture of gas, benzine or petroleum, water and air for the charge.* The heat necessary to vaporize the water keeps the charge during compression at a comparatively low temperature, consequently higher compression pressures can be used without pre-ignition. In the article mentioned the volumes of cylinder and clearance given furnish a value of r equal toabout 10, $\frac{K_p}{K_v}$ is given as 1.4. From these values the cyclic efficiency is about .600. The maximum explosion pressure for full load is given at about 655 pounds absolute.



The compression ratio of the Diesel cycle for this maximum pressure would have been about 16, and for normal load, the cyclic efficiency is about .565. This shows clearly that the superiority of the Diesel cycle may be easily overcome by

* See "Sibley Journal," April, 1903.

slight changes in the conditions. The pressure at the end of compression for the Banki Motor was about 250 pounds absolute, for the Otto cycle, $r' = 6$ in the diagram, it is about 172 pounds absolute. These can not be directly compared with absolute accuracy, since $\frac{K_p}{K_v}$ is not the same in the two cases, but the example shows what a decided change an increase of about 80 pounds in the pressure at end of compression produces.

In conclusion, an ingenious example of how theoretical considerations will sometimes furnish the key to the causes of variations in results obtained on actual tests was given by Prof. E. Meyer, in the "Zeitschrift des Vereins Deutscher Ingenieure," May 9, 1903. Tests on a 14-H.P. Marienfelde alcohol motor, and a 70-H.P. Diesel motor gave the following results:

THERMAL EFFICIENCY ON BRAKE H.P. LOAD.

	Full	*Normal*	¾	½
70-H.P. Diesel motor, per cent.....	32.6	31.9	30.5	27.4
14-H.P. Marienfelde motor, per cent.....	32.7	29.0	...	22.7

It will be noted that the efficiency is much better sustained throughout the range of load in the Diesel than in alcohol motor. Meyer gave the explanation for this on theoretical grounds as follows:

Equal friction losses are assumed. In the figure the full line represents the full-load Diesel card, the dashed line the half-load Diesel card, and the dot-and-dash line the card of the alcohol motor. Combustion in the Diesel engine is assumed to be along the constant-pressure line *a b*. The compression ratio $r = \frac{V_0}{V_c'} = 15.40$ for the Diesel, and $r' = \frac{V_0}{V_c''} = 10.26$ for the alcohol motor. Pressure at end of compression is 510 pounds absolute in the Diesel, and 240 pounds absolute in the alcohol engine. Line *a b* is found to be .09 V_0, and line *a c*, for half load, = .04 V_0. A vertical line drawn at V_c'', cuts line *a b* at *d*, inside of point *c*.

Suppose each diagram is divided into a number of infinitesimal elementary diagrams by a number of adiabatics. It will at once be seen that $r' = 10.26$ for each one of the elementary cycles of the diagram from the alcohol engine. From the formula for E, all of these elementary cycles have therefore the same cyclic efficiency. In the case of the Diesel full-load card, all the elementary cycles to the left of d have a ratio of compression greater than 10.26, and these are therefore of higher efficiency than the elementary cycles of the alcohol motor. But all those lying to the right of point d have a value of r constantly decreasing as we pass further to the right toward b, and the cyclic efficiency therefore constantly decreases. The average of the efficiencies obtained for all the elementary cycles of the full-load Diesel card will therefore just about equal the efficiency of the alcohol diagram, because c and d are close together, and hence we find the thermal efficiencies of the two engines actually about equal at full load. For half load, however, the case is different. The alcohol-motor card does not change sensibly owing to the method of regulation. But the average of the elementary Diesel cycle efficiencies is now greater than that of the alcohol-motor cards because but few elementary Diesel cycles lie to the right of d. Hence nearly all of them have a value of r greater than 10.26, and hence the cyclic efficiency is greater on the average. We would therefore expect a higher thermal efficiency at half load from the Diesel than from the alcohol motor, which is what actually occurs as shown on test.

II. FOUR-CYCLE *vs.* TWO-CYCLE ENGINES.

The development of the gas engine, more particularly of the high-powered engine, has made the question of 4- *vs.* 2-cycle engines of the highest importance. The literature on this subject is not at all extensive, and what there is does not seem to bring the controversy to any definite conclusion. The main reason for this state of things is, perhaps, the absence of very conclusive tests, for it cannot be denied that however

much a subject may be discussed theoretically, practical tests, if they can be made, will in all cases decide the question.

The history of the subject is in brief as follows. The four-stroke cycle is undoubtedly the idea of Beau de Rochas, who fully explained its action in a pamphlet of 1862. About the same time Otto attempted to make the 4-cycle engine a practical success. Unable, however, to overcome the difficulty caused by the severe blows of the exploding gas mixtures, he in 1862 gave up the development of the direct-connected 4-cycle engine, and for the next fifteen years, in company with Langen, developed the free piston engine. This engine won them the first prize at the second Paris Exposition. Apparently about 1875 Otto returned to his original idea, and his success along this line was well shown at the Paris Exposition of 1878. Among 75 machines there exhibited, the Otto 4-cycle engine was distinguished above all others both by quietness of operation and by greater power per cubic foot of cylinder volume.

From 1874 to 1886 the Gas Motoren Fabrik Deutz monopolized the construction of 4-cycle engines, under the protection of German Patent No. 532. After a legal fight, lasting nearly four years and going on at the same time in Germany, Austria, England, Belgium, France and Italy, the scope of the patent was considerably reduced, Claim No. 4, covering the 4-cycle principle, the especial bone of contention, being vacated. In the meantime, under the force of circumstances, the other manufacturers were compelled to develop the 2-cycle engine, and there is no doubt that if the state of affairs as outlined had continued to the end of the patent, the 2-cycle engine would have reached its present development much earlier. The very fact, however, that the 2-cycle principal in its infancy offered numerous and serious difficulties, made the legal fight protracted and bitter, the above mentioned pamphlet of Beau de Rochas, of 1862, being a very effective weapon against the validity of Claim No. 4. The natural result of the vacation of this claim in 1886 was that the 2-cycle machine almost disappeared from the market for

nearly 10 years, and has found its present development within the last 6 or 8 years.

As stated at the outset, the question of choice between the 4-cycle and the 2-cycle machine as regards economy, operation and maintenance, is not at all decided, although it must be admitted that the decision commences to lean more and more strongly in favor of the 2-cycle. Güldner, especially, to whom we shall be indebted for a great deal of what follows, decides very strongly in favor of the 2-cycle machine. Other authorities are not so decided. There is a great deal that may be said for and against each type on both theoretical and practical grounds. Before comparing the two methods as to efficiency, it will be well to outline the conditions attending each.

The two types are thermodynamically equal as far as the combustion of the charge, whether at constant volume or constant pressure, and their expansion strokes are concerned. They differ only in their method of displacing the old and taking in the new charge, *i. e.*, in their scavenging and loading.

In the 4-cycle machine the scavenging is done by the power piston pushing out the burned gases through the exhaust port. This is followed by the suction stroke, during which the new charge is taken in. The compression stroke, combustion, and expansion stroke following complete the cycle. The piston therefore works half the time as a power piston, and the remainder of the time as a pump piston.

In the 2-cycle machine the scavenging and loading are done in various ways. In the first place we may distinguish two scavenging agents: the fuel mixture and air alone. Further the scavenging agent is introduced into the cylinder by pumps in various ways. Among these we may distinguish three types: the enclosed-crank case employed as a pump, the front end of the cylinder employed as a pump, and a pump entirely independent of crank case or power cylinder. A further modification comes in when any of the above types of pumps are used with or without an air receiver. Several methods of scavenging are also employed. The fuel mixture alone may

be used to drive out the exhaust gases, or a little air may be sent into the cylinder ahead of the fresh gas, or, finally, the scavenging may be done by an excess of air, followed by the admission of fuel mixture when the scavenging is complete. How these various methods compare among themselves, and with the 4-cycle, will be seen later on.

The final judgment upon the success or non-success of an engine should always be based upon the thermal efficiency of the machine at the crank shaft and upon the mechanical efficiency. Of the power generated during the working strokes of the cycles a part is lost in the fluid friction of the machine and in the rubbing friction of the various machine parts. The fluid friction in a 4-cycle machine will be understood to mean the pump work as indicated by the bottom loops of the indicator cards. In careful testing these loops should always be taken with a weak spring, in order to determine the pump work with more accuracy. This is especially desirable since little data exists on this point. Should the 4-cycle engine be of the positive scavenger type, as the 500-H.P. Premier reported upon by Humphrey in 1900, this pump work should be added to the fluid friction of the bottom loops. In a 2-cycle engine the fluid friction is the pump work done by the fuel and air pumps. The rubbing friction of the machine need not be further defined.

It seems that engineers are not quite unanimous in their methods of computing mechanical efficiency, as the writer had occasion to notice lately in looking over some tests. If from the total indicated horsepower we subtract the fluid friction, in either type of engine, we shall obtain the net indicated horsepower. The mechanical efficiency is, according to some,

$$E = \frac{\text{B.H.P.}}{\text{total I.H.P.}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

according to others,

$$E = \frac{\text{B.H.P.}}{\text{net I.H.P.}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Güldner consistently employs formula (2), which gives a higher result than formula (1), and takes pains to correct

Thurston's figures on a Brayton gas engine, computed according to (1). It seems to the writer that the use of formula (2) is justified only when the work of the air pump is to a great extent recovered in the power cylinder, as is the case in a Brayton engine, or in the Diesel. When the air compressed by the pump is, however, used for scavenging only, blowing through the power cylinder when the exhaust port is open, this is not the case, and for such engines, the majority of 2-cycle engines, it seems nearer right to use formula (1). Similarly the thermal efficiency at the brake should be

$$E' = \frac{\text{Thermal equivalent of B.H.P.}}{\text{Thermal Units in Fuel supplied}}.$$

A complete comparison of the 4-cycle and 2-cycle principles should include the following heads:

I. Thermodynamic actions in the cylinder.
II. Fluid friction.
III. Friction of the machine.
IV. Limitations of construction and economic considerations.

I. THERMODYNAMIC ACTIONS IN THE CYLINDER.

Thermodynamically the methods are equal, the utilization of the heat in practice is, however, influenced by several extra-thermodynamic actions. In the first place, the cylinder volume of most 2-cycle engines is much smaller, approximately about one-half, than the cylinder volume of a 4-cycle machine of the same power. The enveloping cylinder surface is therefore greater per unit volume in the 2-cycle than in the 4-cycle engine. This means that more heat will be carried off by the cooling water during the compression, and the higher compression pressure that this fact permits of carrying means finally an increase in the thermal efficiency. Of course the increased loss of heat also takes place during combustion and expansion, affecting the results unfavorably. It seems, however, that the somewhat lower average temperature of the highly-compressed lean mixtures which may be used in the 2-cycle engine about evened up this loss. As a matter of fact, tests have shown

that the proportionate heat loss to the jacket water does not differ much for the two methods, varying only with size of engine—sometimes in favor of one, sometimes of the other—of the two methods.

Thus it seems that the 2-cycle engine may use a slightly higher compression pressure than the 4-cycle for the same mixture, with its attendant favorable result. This advantage is further increased by the fact, as pointed out by Güldner, that the 2-cycle, if properly handled, works with a lower temperature at the beginning of compression than the 4-cycle, at least in an engine employing air for scavenging, since the scavenging action takes place through only about $\frac{1}{3}$ of the stroke as a maximum, allowing but little time for the taking up of heat from the walls. Further, the scavenging air is usually under a pressure slightly greater than atmospheric, and this and the lower temperature mean finally a greater weight of charge per cycle and consequently greater specific power. In the 4-cycle the loading takes place under suction, lasts during the entire stroke, the charge has had considerable time to warm up, and as a result we have a higher temperature and a lower pressure at beginning of compression than in the 2-cycle machine. Both of these tend to lower the pressure realized at end of compression, the effect of the lower initial pressure being obvious, and that of the higher initial temperature serving to decrease the weight of the charge. Besides, depending of course on the nature of the fuel used, a high initial temperature may set a low limit to final compression pressures because premature explosions must be avoided. Without reference to the effect of temperature, if we assume the pressure of the scavenging air after scavenging to be about 15 pounds absolute, that at the end of the suction stroke in a 4-cycle machine to be about 14.0 pounds, the weight of charge in the 2-cycle will be $\frac{15-14}{14}$ = about 7 per cent. greater than in the 4-cycle machine. We gain about that much in specific power, *i. e.*, power developed per cubic foot of piston displacement, because, other things being equal, the heat converted into external work is directly proportional to the weight of the charge.

Summarizing the above we have the following:

1. Lower temperature and higher pressure of charge at beginning of compression in the 2-cycle, resulting in

2. Greater weight of charge and consequent greater specific power.

3. Greater loss of heat to cooling water during compression, allowing of higher compression pressures, and resulting in

4. Higher thermal efficiency.

Tending to show the correctness of the above discussion, Güldner cites the case of the Diesel-Güldner 2-cycle engine, in which the pressure at end of compression had to be made from 60 to 90 pounds higher than in 4-cycle machines, in order to attain temperatures high enough to ignite the charge.

II. FLUID FRICTION.

The pump work in a 4-cycle machine is done in the power cylinder itself. The disadvantages attending this method have in part been already pointed out. The efficiency of the power cylinder as a pump as compared to an independent pump must be low, because, to state it briefly, the power cylinder is not designed as a pump. The valves are not designed as pump valves, being designed to stand the much higher pressures of the power strokes, and the large clearance spaces and high temperature of cylinder walls affects very seriously the capacity of the cylinder as a pump. It may be objected that the fluid friction in a well-designed 4-cycle machine is not a very serious loss, as may be judged from mechanical efficiencies obtained, but it can not be denied that if the same work were assigned to a cylinder designed as a pump, with proper ports and valves and cool cylinder walls, the same work would be performed at a still lower cost. Besides, as Humphrey pointed out in a test on a Crossley 400-H.P. 4-cycle engine, not only the length, size and arrangement of piping affect the fluid friction very seriously, but the load on the engine also changes it. The amount of this change depends, however, on the kind of fuel used and on the system of governing. Humphrey found that when only air was taken in

the bottom loop cards indicated a fluid loss of about 33 H.P. at normal speed. When exploding every cycle this loss was about 15 H.P. The fluid loss at full load is therefore about $\frac{15}{400} = 3.8$ per cent. Even if we assume that the fluid loss at half load is not at all increased, which can not, however, be fairly expected on the basis of the above figures, the fluid loss would have increased to $\frac{15}{200} = 7.5$ per cent. And with a loss of from 20 to 22 H.P., which is probably nearer the truth, it would have risen to from 10 to 11 per cent.

Finally, unless the engine is of the positive scavenging type, the scavenging in a 4-cycle engine cannot be perfect, since the combustion chamber is always left full of the burned gases. This is unavoidable, and its most serious effect is to decrease the weight of charge in the cylinder, both by its own bulk and by unduly heating the incoming fresh gases by mixing with them.

It is evident from the above that the pump work if performed by the power cylinder can not be done to best advantage. Cutting out the two pump strokes at once converts the machine into a 2-cycle engine, and we have next to examine how the case stands when the pump work is performed in a pump other than the power cylinder. As pointed out above, an examination of existing types of 2-cycle engines shows that this work is done in several ways, some of these ways being undoubtedly responsible for the present bad repute which the 2-cycle machine now enjoys in some quarters. Before deciding between the various means employed it will be well to outline the conditions that attend the scavenging and reloading of the cylinder of a 2-cycle machine.

At the end of the expansion stroke in the 2-cycle machine the cylinder is filled with burned gases at approximately atmospheric pressure. These gases must be replaced to give room for the next charge, and upon the perfection of scavenging depends the capacity, economy and satisfactory working of a 2-cycle engine. This is, in fact, the critical phase of the 2-cycle principle, and it is not too much to say that, depending upon the perfection of scavenging, the 2-cycle will stand or fall.

While the scavenging of the 4-cycle engine is mechanically a simple process, the same cannot be said for the 2-cycle engine. The action going on during the scavenging of a 2-cycle machine is very complex, mainly on account of the time allowed for the process. While the theoretical aspects of the process are very clear, the practical side is not so simple nor so well known. The reason for this is probably that manufacturers of 2-cycle machines are rather backward about giving out their good or bad experiences. Another reason may be, as Güldner somewhat sarcastically remarks, that "to science and to the mechanical laboratories of our technical institutions the 2-cycle principle does not seem to offer any further questions worthy of investigation."

The ideal scavenging process in a 2-cycle engine would be the following: scavenging should commence when the pressure of the burned gases has fallen to nearly atmospheric pressure. In order to insure a rapid establishment of equilibrium after the exhaust port has opened, this port should be a ring of openings in the cylinder wall instead of a valve or opening of the ordinary type. Scavenging should, therefore, commence before the piston has reached its outer dead center. It is of extreme importance, however, and for obvious reasons, that the pressure of the burned gases be less than that of the scavenging agent, and it is better, therefore, to be a little too late than too early with the admission. The scavenging agent should be under low pressure. High pressure causes it to flow into the cylinder under high velocities, besides unnecessarily increasing fluid friction. It is apt to pierce the burned gases, strike the cylinder walls, and rebound from them. This causes eddy currents and sometimes results in a very thorough mixing up of burned and incoming gases, which is just what is to be avoided. If, on the other hand, low pressures be employed, the incoming gases enter quietly, and if the ports have been properly designed, they will fill the cylinder from wall to wall in a solid cylinder, pushing the burned gases out ahead of them. Low pressure and properly-placed inlet ports are therefore essential. This, on the one hand,

again emphasizes the importance of delaying the admission of fresh gas till near the end of the outstroke, and, on the other hand, points out the necessity of careful design of combustion chamber and ports. As a last requirement an excess of the scavenging agent under as nearly as possible constant pressure should be used, for only in this way can thorough scavenging be assured, and this means, indirectly, that scavenging should continue as long as possible, *i. e.*, to the closing of the exhaust ports on the return stroke. It is true that the beginning of compression is thereby delayed until from 10 to 12 per cent. of the return stroke is completed, but the greater weight of charge present in the cylinder, as compared to the 4-cycle, evens up this loss.

Summarizing, we shall therefore have for ideal conditions of scavenging in the 2-cycle engine:

1. Commence scavenging near the dead center on the out stroke, and continue to the closing of the exhaust port in the return stroke.

2. Use scavenging agent under low pressure, high pressure resulting in failure. At present the pressures range from 5 to 10 pounds by gauge. Have properly designed combustion chamber and inlet ports, to prevent as far as possible any breaking up or shattering of the incoming gases. Failure in this will mean imperfect scavenging.

3. Use an excess of the scavenging agent, and maintain as nearly as possible a constant pressure during the operation.

In the light of the above conditions, we are now able to compare the methods of scavenging employed in existing 2-cycle engines.

Referring to the scavenging agents used, and the methods of using them, the engines employing fuel mixture alone are under a great disadvantage because no excess of the scavenger can be used since this would result in a direct loss of fuel through the exhaust port. Imperfect scavenging is the inevitable result, and since it is next to impossible to prevent the loss of some fuel through the exhaust port, such engines can have no claim to economy. Where the crank case is used for

compressing the mixture a further fuel loss is apt to result through leaky frame and bearing joints. Besides this, the crank-case pump is very inefficient owing to large clearance spaces, and it possesses the further disadvantage that scavenging ceases as soon as the piston starts on the return stroke, thus losing a good part of the scavenging period. In view of all these disadvantages we find that this method of scavenging and loading is confined to small 2-cycle machines in which a low first cost is usually of much more importance than great or even fair economy.

If the ready-fuel mixture is sent into the power cylinder by an independent pump, the case is much better as far as pump efficiency is concerned, but the fuel loss through the exhaust port, which is the most important part, still exists.

The above considerations point to the conclusion, first, that a crank-case pump cannot be expected to give satisfactory results, and, second, that, except in such cases where economy is of secondary importance, air should be used for scavenging, either wholly or in part.

Concerning the choice of pumps, an independent air pump with an ample air receiver furnishes the ideal conditions as above outlined, for, under these conditions, we can commence or cease scavenging at will, dependent upon the setting of the valves, and, if the receiver is large enough, an excess of air under fairly constant pressure is available. If the receiver be made too small, scavenging will cease too early, an excess of air not being available, and the scavenging pressure decreases very rapidly. Cutting out the receiver altogether, and discharging from the pump directly into the power cylinder, makes the conditions much less favorable, for, if the main crank and the pump crank are 180 degrees apart, scavenging ceases when the main piston has reached its outer dead center, and if the pump crank be made to lag or lead a few degrees, as the case may be, the scavenging pressures are apt to be unduly high, and they decrease very fast. In the third type of pump, employing the front end of the power cylinder, we have two other points entering. One is that no excess of air

is available, the other, that the capacity of the pump is further decreased by the unavoidable heating of the air during suction by contact with the cylinder walls. These considerations lead to the following ranking of the types of pumps in order of merit.

1. Independent pump with ample receiver.
2. Independent pump without receiver.
3. One end of power cylinder employed as pump cylinder.
4. Crank-case pump.

Whether air entirely or only in part be used for scavenging again depends upon consideration of first cost mainly. The idea of using air in part only is that, if a little air is sent into the cylinder ahead of the fuel mixture, if any of the fresh gases are lost through the exhaust ports, it will be the air alone. This has in some cases been accomplished by letting the fuel pump suck air only during the latter part of its stroke. The air will then be the first to enter the power cylinder. This method cuts out the air pump, and has this fact in its favor. In other cases, as in the Koerting, the air pump fills the pipes leading from the gas pump to the cylinder with air, so that as the gas pump starts to deliver, it first presses this air into the power cylinder. The success in either case depends upon the assumption that the three layers of gas at one time in the power cylinder, *i. e.*, burned gas, air and fuel mixture, remain separate during the scavenging. Any production of eddy currents in the cylinder will defeat the design, and it can at once be seen that failure is much more likely to result than success. The use of air alone for this work therefore seems to offer the best chance of accomplishing the desired end.

The following cards taken from air receivers and crank cases show the action in each during scavenging.*

Fig. 1 is taken from an air receiver which was too small. The pump delivers air into the receiver from *a* to *b*. The maximum pressure is about 5 pounds, and this remains constant to *c*. At *c* scavenging commences, and the pressure falls

*Güldner Entwerfen und Berechnen der Verbrennungsmotoren, pp. 186, 187.

rapidly to about 1.5 pounds at *d*, near the end of the power piston stroke; after that very little air flows from the receiver, and the best part of the scavenging stroke, from *d* to the closing of the exhaust port, has not been made use of.

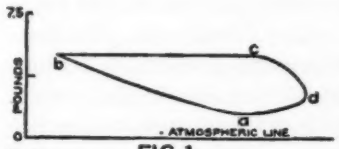


FIG. 1

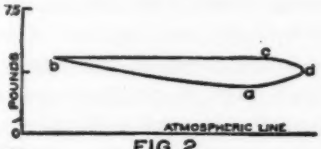


FIG. 2

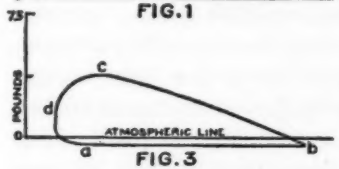


FIG. 3

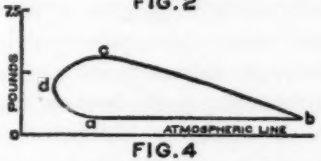


FIG. 4

Fig. 2 shows a card taken when the receiver volume was considerably increased. Here the maximum pressure is not quite so high, but scavenging is continued from *c* through *d* to *a*. The pressure drops only to about 3 pounds, and since this is the pressure also existing in the cylinder at the close of the exhaust and air ports, more weight of charge is present in the cylinder, increasing the capacity of the machine.

Fig. 3 shows what takes place in the crank case. Suction occurs from *a* to *b*, compression from *b* to *c*, at *c* air is admitted to the power cylinder, and the pressure rapidly drops to *d*. Here the return stroke commences, scavenging ceases, and the pressure rapidly drops to suction pressure.

In Fig. 4 the conditions of Fig. 3 have been improved by making the pipe leading from the crank case to the cylinder as large as the space would allow and fitting this pipe with a check valve, thus creating a limited receiver. This bettered the conditions very much, scavenging taking place from *c* through *d* to *a*, continuing after the piston has commenced its return stroke, owing to the action of the check valve. Only in this way is it possible in crank-case scavenging pumps to attain anything like favorable results.

From the foregoing it is clear that if the 2-cycle engine is

to compete successfully with a 4-cycle machine, the scavenging must be done carefully and thoroughly, and, in order to do this, independent pumps and ample air receivers are essential. Every other means is but a compromise, and should come under consideration only where first cost is the main factor, and economy but of secondary importance. Further, it would seem on the purely theoretical grounds so far outlined, that to do this work should cost less in the 2-cycle than in the 4-cycle machine. It is not easy to show whether this is true in practice or not from existing data. Recorded tests usually are incomplete in that they give the difference between indicated and developed horsepower only without differentiating the friction loss into fluid friction and friction of the machine. Güldner feels nearly sure that the fluid loss in the 2-cycle can be brought down to or be made less than that in the 4-cycle. According to him, this loss in the 2-cycle machine is at present about 8 to 10 per cent. of the indicated work, as compared to from 6 to 7 per cent. in the 4-cycle, but he feels very certain that as the 2-cycle type is improved this difference of from 2 to 3 per cent. will ultimately disappear. Schöttler, in the "Zeitschrift des Vereins Deutcher Ingenieure," Oct. 4, 1902, speaking of the engines exhibited at Düsseldorf in 1902, says: "The 2-cycle has the unquestioned advantage that but half the cylinders are required (for same power). This is negatived to some extent by the fact that pumps are necessary, but these will take more power than the pump work in the 4-cycle." No figures are given to support this statement, but Güldner agrees with him in so far as he admits that the 4-cycle has a slight advantage today in respect to fluid friction. This, however, may merely show that the practice of today has not been able to fully realize the advantages offered by the 2-cycle machine, which is not at all surprising when the period of development is considered. The conclusion would be that the difference, if any, is as yet in favor of the 4-cycle as regards fluid friction, but that as experience teaches and practice improves, this difference will disappear.

III. FRICTION OF THE MACHINE.

Turning next to the friction of the machine, the points to be considered here are the friction of pistons, crossheads and bearings. If we consider that the rubbing friction in an engine is in direct proportion to the volume of the cylinder per unit of power, then, if the pump volumes are added to the cylinder volume in the 2-cycle, the two methods would seem to be about on a par. The pump piston in a 2-cycle machine, however, works under comparatively low pressures, and the rubbing surfaces are cool and well lubricated. The pressures being low, the piston rings may be given an easy fit, and, all these things considered, the friction in the pump should be low. In the 4-cycle engine we have a machine member, the piston, designed to stand pressures of from 250 to 450 pounds, working half the time under pressures of from 14 to 16 pounds, a state of things not conducive to best results. Under such circumstances it is hardly necessary to state on which side the advantage as to piston friction lies. Nearly all of the smaller-size motors use a trunk piston, working without a crosshead. This is done in both 2-cycle and 4-cycle machines. The practice can not be called a good one, because it makes the work of the piston very severe, using highly heated rubbing surfaces to take up transverse stresses. If a crosshead is used, there is no good reason why a difference in the rubbing friction in this member should exist when employed in either type of machine. In the 2-cycle machine a crosshead possesses the double advantage that it relieves the working piston from other than longitudinal stresses, and it may be designed to serve as a piston for the pump. In regard to bearing friction, the 2-cycle machine, with a smaller piston for the same capacity, has somewhat smaller dimensions in bearings and journals, and hence the friction loss is somewhat less at these points. The fact that an impulse is given every revolution also allows of a smaller weight of fly wheel for the same requirements of regulation, and this, reducing the pressure on the bearings, of course serves to further decrease the friction loss. All of this finally seems to point to a lower total friction loss in the 2-cycle engine.

As already pointed out, the conclusion under II and III cannot at the present time be substantiated from the existing data. Of 4-cycle tests there are a great many, but the number of 2-cycle tests is very small. The practical proof is, of course, a demonstration of equality as regards mechanical efficiency. This proof can not at present be brought. The writer has collected all the available tests from a great variety of sources, but while they show that the *best* figures are obtained by the 4-cycle engine, it is by no means clear that the *average* figures for the 4-cycle are better than those for the 2-cycle.

IV. LIMITATIONS OF CONSTRUCTION AND ECONOMIC CONSIDERATION.

While the considerations under II and III seem to point out that the 2-cycle is but little behind the 4-cycle engine as regards power losses in the machine, and that what little advantage the 4-cycle now possesses will probably disappear as soon as the conditions governing fluid loss are better understood, it must not be concluded that it is therefore immaterial in any given case whether a 2- or a 4-cycle machine be used. Questions of construction and of economy enter the problem, and these will usually have an important bearing upon the final choice.

According to speed and power, gas engines may be roughly divided into the following three classes:

1. Speeds in the neighborhood and above 500 r.p.m., power comparatively low, not to exceed 20 to 30 H.P., as a maximum. Such engines are used in the automobile industry, for marine launches, etc.

2. Speeds from 200 to 500 r.p.m., power up to about 100 H.P. This class includes the engines used for general all-round power work and production of electricity.

3. Speeds below 200 r.p.m., and power up to 500 H.P. and over. These engines have as yet limited application, being used mainly in connection with blast-furnace gas, for the generation of current and production of blast for the furnaces.

1. The engines used for automobile work are nearly all 4-cycle machines. The same may be said of launch engines,

although many 2-cycle machines are also used for marine work. The use of 4-cycle machines for these purposes is easily explained on several grounds. In the first place economy is of secondary importance in nearly all of these cases. The first consideration, except perhaps for marine work, is lightness combined with power. This at first sight would lead to the choice of a 2-cycle machine, as having a higher power per cubic foot of cylinder volume. The high speeds found, however, in this work make a successful operation of the 2-cycle principle very uncertain because the time allowed for the various events is exceedingly small. The development has therefore been along the lines of 4-cycle machines with very high speeds, to attain power combined with lightness. The results attained are quite remarkable, especially as regards motor for air ships. The firms mainly concerned in this are Daimler, DeDion and Bouton, and Buchet.* In the last half of the 90's, Daimler built for Dr. Wölfert a 6-H.P. motor weighing about 90 pounds per H.P. This was a 4-cylinder machine making 800 r.p.m. Daimler also built the motor used by Von Zeppelin in his remarkable work. Two motors were used, each of 12 nominal H.P. at 700 r.p.m., which could be increased to 16 H.P. for a short time. These machines were an improvement over that used by Wölfert. The weight of each, including cooling water, benzine and accessories, was about 1,000 pounds, which, at 16-H.P. would be about 64 pounds per H.P. Since that time DeDion and Bouton have, by increasing the revolution to about 2,000 per minute and by the use of valves of large cross-section, reduced these figures considerably, although, of course, at the expense of economy. Buchet worked along the same lines. For speeds of about 2,000 r.p.m. Buchet gives the following figures for his automobile machines, the weights are exclusive of vaporizer, benzine, and cooling water:

Power.	No. of Cylinders.	Total Weight.	Weight per H.P.
8 H.P.	2	114 pounds.	14
16 H.P.	4	202 pounds.	13
24 H.P.	4	238 pounds.	10

*Von Paller, Z. d. I., Aug. 16, 1902.

Both Santos-Dumont, in Paris, and Kress, in Vienna, used Buchet motors in their later experiments. Although figures are not available, the writer has been credibly informed that the above figures have been reduced.

2. The choice of the type of machine for general power work is somewhat dependent upon the purpose for which it is to be used. The factors entering are not only economy of fuel, but also first cost, maintenance, certainty of operation and regulation. Admitting that the 2-cycle machine can not at present quite show the economy of the 4-cycle engine, it has the advantage in all of the other items named, with the possible exception of certainty of operation. It may be said, however, in regard to this latter point, that unreliability of operation was associated with gas engines in general in the minds of the public for a long time. Later on the charge was made against 2-cycle machines only. In neither case is it at the present time justified, and in no case except in small very high-speed machines, for reasons above pointed out, does the 2-cycle machine give any trouble in this regard. If the engine is to be employed for the driving of generators, the regulation is of the highest importance. Two-cycle machines have here an unquestioned advantage because of much more uniform crank effort with a lighter fly wheel. The same thing can be obtained by using two cylinders in the 4-cycle type. This, however, camplicates the machine, and merely emphasizes the advantage of the 2-cycle as to first cost and maintenance.

3. Schöttler, in the "Zeitschrift" for June 14, 1902, says, with reference to the engines exhibited at Düsseldorf:

"To make an attempt at a decision as to which one of the two working cycles, now employed in large gas engines, belongs the future, or, to put it more modestly, from which one of the types of engines exhibited will develop the type of the future, would be too early. They all have their advantages and their faults. That the theoretical man leans toward the 2-cycle, and the practical man toward the 4-cycle, is easily understood. It was quite natural, when the iron and

steel industry demanded large gas engines, that an attempt should first have been made to adapt the tried 4-cycle principle. . . . But that the 2-cycle should soon be tried was also not to be wondered at since smaller dimensions could probably be attained. Theoretically it is immaterial as regards regulation whether four 4-cycle cylinders or one double-acting 2-cycle cylinder be used; in either case lighter fly wheels may be used than if a smaller number of cylinders and single action had been employed."

Schöttler also gives the following table of the details of some large gas engines in the same article.

	Brake H.P.	Cylinder.		Revolutions per minute.	Piston speed, feet per minute.	Piston displacement for 1 H.P., cu. ft.	Weight, tons, metric.		Coef. of regulation.
		Diameter, inches.	Stroke, inches.				With fly wheel.	Without fly wheel.	
Single-cylinder 4-cycle Société Cockerill.....	575	51.2	55.2	93	860	.115	127	94	1 : 20
4-cylinder 4-cycle Deutz.	600	26.0	33.6	150	840	.068	106	88	1 : 130
Single-cylinder double-acting 2-cycle Körting.	500	25.0	43.4	100	724	.025	70	58	1 : 80

H. Neuman, in a discussion on Meyer's report on a test of a Körting engine, stated that the Deutz company were not building 2-cycle engines, because computations had shown, allowing an indicated pump resistance of 6 pounds, that the mechanical efficiency of a 2-cycle machine was about 74 per cent., while that of a 4-cycle, under the same conditions, was nearly 90 per cent. The economy, everything else being the same, would be in the ratio of the efficiencies. That is a strong statement against the 2-cycle machine, and it would be interesting to know whether practical experience has corroborated these computations. So far nothing conclusive has been published to prove the point.

Agreeing with Schöttler that it is theoretically immaterial

as regards regulation whether four 4-cycle cylinders are employed, or one double-acting 2-cycle cylinder, it is not quite so immaterial in many other respects. Comparing the Deutz and the Körting engines, from the above table we find the piston displacement for B.H.P. in the Körting 2-cycle machine is only 37 per cent. of that in the Deutz 4-cycle engine. The weights per H.P. compare as follows:

	Deutz	*Körting*	*Per cent.* $\frac{Körting}{Deutz}$
Gross weight per H.P., tons.....	.17	.14	82
Without fly wheel, weight per H.P.	.15	.12	80

This shows a gain of about 20 per cent. in favor of the 2-cycle engine. This, combined with the fact that four cylinders must make an engine much more complicated and more difficult to handle than a single cylinder with pumps, gives a very strong argument for the 2-cycle in large engines.

The co-efficient of regulation, *i. e.*, $\frac{\text{max. speed} - \text{min. speed}}{\text{average speed}}$ is better in the Deutz engine, but it is bought at the expense of a fly wheel 50 per cent. heavier. This not only serves to increase first cost, but tends to decrease mechanical efficiency. For ordinary purposes the co-efficient $\frac{1}{80}$ is sufficiently low to come within specifications.

In conclusion, there is one aspect of the case which grows in importance as the demand for greater power from least expenditure becomes more imperative. More than 500 H.P. have been produced in a single 4-cycle cylinder, but beyond that the difficulty increases very seriously. The large dimensions necessary make very costly construction, and the difficulty of obtaining a proper mixture of fuel and air in such large cylinders, combined with consequent uncertainty of operation and loss of efficiency, soon sets a practical limit to the power generated in one cylinder. For very large machines it is therefore no longer a question of economy only, but a question of possibility of construction and of operation, and, unless recourse be had to complicated multi-cylinder machines, the solution is obvious.

III. MATTERS OF DESIGN, AND THE PRINCIPAL DIMENSIONS.

It may be stated with some satisfaction that the era of hunting after novel constructions, *i. e.*, something not before accomplished, is about passed in gas engine design, and that the subject is being gradually but surely placed on a common sense business footing. Time was when nearly every independent designer was bound to produce something distinctively his own, and this tendency led to many curious and sometimes absurd constructions. Happily it is becoming more and more widely recognized that the underlying principle that should guide all design, *i. e.*, best adaptation of the means at hand to the particular circumstances of every different case, applies with as great force to the subject of gas engine design as it does to the other fields of the mechanical engineer's activity.

In gaining an idea of the true economy of an engine, we must take into consideration all of the costs of operation. These not only include the cost of fuel and of maintenance, but also interest on capital invested, and depreciation. The capital invested depends directly upon the market price, and this in turn upon the manufacturing cost of the machines. In many cases the interest on the invested capital is a large part of the yearly operating cost, and for this reason, and on account of the fact that a low market price will increase the selling qualities of the machine, the designer must always keep in mind manufacturing costs.

It is hardly necessary to point out that the various types of motors should each have its separate treatment in design. The requirements of a marine motor differ from those of an automobile engine, and, in general, the design of small motors should be governed by somewhat different views than that of a large engine. Broadly speaking, a small engine should be designed with a view to the greatest possible simplicity, a large engine with a view to obtaining the highest possible economy. These should be in each case the leading ideas. In the first case, however, economy must not be entirely lost sight of, nor simplicity of construction in the second. The possession of proper judgment as to how far the primary considerations may

be deviated from in favor of the secondary, makes the sucessful designer. One thing must never be forgotten. In all cases reliability of operation and durability are of the utmost importance. One hour of interruption in an important manufacturing plant may swallow up all the saving in fuel of a year, and an engine with the very highest thermal efficiency is but a poor investment if it has to be scrapped at the end of a few years.

There are several minor points which should also receive due consideration. The maximum pressures occurring in gas engines are from 3 to 4 times those occurring in steam engines, and the time of their occurrence is not exactly the same, turn after turn. The engine should therefore be so designed that the stresses may be taken up centrally as nearly as possible. For this reason the Tangye type of frame has not found much application in gas engine design. The reason is that the crank transmits the pressure through a moment arm to the bearing, producing a bending moment on the frame, which, in view of the high pressures occurring, is very severe. Again, one bearing is required to carry the total pressure. Both of these things require a much heavier frame of this type in the case of the gas engine than in that of the steam engine. While steam engine designers have found this frame well adapted to even the highest powers, the altered conditions of gas-engine practice do not allow of its use, at least for large engines. Design the frame so that all stresses may be easily computed and checked. This will in itself lead to simplicity of form, which is conducive to safety and economy, and while this is to be commended it is well to guard against the other extreme, plainness degenerating into ugliness.

The whole arrangement of engine and accessories should be as simple as possible. A complicated layout of valves, cocks and handles merely tends to confusion. A printed set of instructions, if at all elaborate, usually fails to fulfill its purpose, being either never read, or, if read, usually misunderstood. Parts of engines which are set at the factory, and which should not, or can not, be moved by the engine tender without en-

dangering the safety of the machine, had best be fixed rigidly in position, as many attendants have an inquisitive turn of mind.

Among the points of design still discussed in the literature on the subject are the following: Vertical or horizontal engines, the use of the crosshead, single- or double-acting cylinders, multicylinder construction, and compounding. The views for and against these matters will be given as briefly as possible.*

The gas engine of today, outside of the small machines, is of the horizontal type, the most notable exception being the Westinghouse in some of its forms. It seems to be assumed that the horizontal is the better form, but the vertical arrangement possesses strong points in its favor. In a vertical machine the wear on the piston is much more evenly distributed, and much less, than in horizontal machines. This, with the fact that lubrication is apt to be more uniform, reduces piston friction to a minimum. It is also possible with proper design to nearly avoid bending moments in the frame, leading to a lighter frame construction. This is important, and explains why Diesel and Bánki adhere to the vertical form. In this form it is easy to multiply the cylinder to attain the highest powers, and since no crossheads need necessarily be used, the machines will not need excessive head room. The foundation under this form is subject to vertical stresses mainly, and can therefore be built smaller and at lower cost than that of a horizontal machine of the same power. All of the valve gearing is above ground, and not partly in pits, as is the case in some horizontal machines. Further, the vertical form allows of nearly free expansion by heat in an axial direction, and when it is considered that heads and pistons can be removed with great ease, it must be admitted that the claims of the vertical machine are very strong.

There are, of course, cases where the vertical machine is out of place. Where blast furnace gas, or any of the industrial gases, are used, dust is apt to cause trouble by lodging on the walls or the piston in vertical machines. In horizontal

*For a very complete discussion see Güldner, Die Verbrennungsmotoren.

machines valve ports and combustion chamber may be so designed that the dust will be swept out by the gases. It is also hard to make a vertical machine double acting on account of difficulty experienced in placing and operating the valves in the lower head. Further objections made to the vertical machine are that the valve gearing is apt to be complex and that the crank shaft is not well under control for oiling and adjustment. The former is merely a matter of proper design, and the latter falls when we consider the successful operation of our large marine engines, which are subject to runs of long duration under conditions much more severe than are found in stationary practice.

Whether or not a crosshead should be employed depends upon the size and type of engine. The trunk piston combines two very important functions. On the one hand it must take up the pressures generated in the cylinder and transmit them without any loss through leakage of gases by the piston. On the other hand it is required to take up the oblique stresses caused by the angularity of the rod. It is nearly always bad to make one machine member serve in two capacities, and when the functions so combined are as dissimilar as they are in this case, trouble may result. Loss of pressure by leakage is prevented by the piston rings, and in order to reduce friction to a minimum and to increase durability, they should have an easy fit and run on well-lubricated surfaces. In its function as a crosshead the trunk piston is during the expansion stroke forced against one side of the cylinder, producing a tendency to displace the rings and to cause unequal wear. It is essential that the unit pressure on the side of the cylinder be kept low enough to insure proper oiling and minimum wear. In the earlier years, when the maximum pressures did not exceed from 150 to 180 pounds, and the mean pressures were 50 to 60 pounds, per square inch, it was quite easy to make the piston long enough to insure low unit pressures. And as a matter of fact, in the small and medium-sized machines of today, pressures and dimensions are such that no trouble

whatever need be experienced from this source. The rings may tend to blow through and wear out a little sooner, but this is overbalanced by the great saving in first cost when the crosshead is dispensed with. With maximum pressures from 400 to 600 pounds, and mean pressures in the neighborhood of 100 pounds, the case is, however, somewhat different in large machines. The great pressures produced by the angularity of the rod and the inertia of the great masses of metal require very large trunk pistons in order to keep unit pressures within the desired limit. The inability to properly lubricate large piston surfaces, in extreme cases 50 to 70 square feet, and consequent excessive wear, soon sets a limit to the possibility for constructing engines over a certain power without a crosshead. The disadvantages are greater first cost of machine, greater length, and perhaps a small loss in mechanical efficiency. All of these objections are, of course, overruled by necessity in the case of very large machines, and in such cases the increased first cost and greater length of machines is of secondary importance, while the loss in efficiency is very small when both piston and crosshead are designed to perform their functions properly.

The earliest gas engines were double acting. Their economy, however, was so low that single-acting cylinders were soon employed in order to allow the air to cool the piston. Single-acting machines seem to be the rule today, and for small and medium-sized machines this construction is best, being cheapest and simplest. It is found that, if higher power is desired in these machines, it is cheaper and better to use two single-acting cylinders than to make the machine double-acting. In large machines the case is again somewhat different. All considerations compel the generation of maximum power from minimum cylinder volume. This means the use of double-acting cylinders. Water cooling will have to be employed for the pistons, but since this is necessary also for large trunk pistons, the inconvenience involved should not be charged solely against the double-acting machine. The serious difficulty of this type is in the successful design and

construction of the cylinder head of the crank end. The question of the stuffing box has been successfully solved since the invention of good metallic packings. But the water-cooled stuffing box leaves little room for the valves, and the many openings in the head are bound to weaken it. The idea presents itself to make the piston operate some of the valves in order to remove them from the head, and this leads to the double-acting 2-cycle engine as exemplified by the Koerting engine.

The main reason for employing several cylinders in steam-engine design is to reduce cylinder condensation to a minimum, and thus to save fuel. This reason does not exist in the case of gas engines, but in its place there are several other very strong ones, and among them may be mentioned the following:

1. In single-cylinder 4-cycle machines crank shaft and bearings have to be designed to stand the maximum pressure, although this occurs but momentarily every fourth stroke. This means an extra expenditure of metal and an increased loss by friction.

2. In order to attain anything like satisfactory regulation the revolving masses have to be made heavy. For the lowest coefficient of regulation $\delta = \frac{1}{20}$, which ordinarily occurs, Güldner has computed that a horizontal 4-cycle machine will need about 110 pounds of metal in the rim of the wheel, per horse power, the peripheral velocity of the rim being about 65 feet per second. For a 500-H.P. machine this would mean a rim weighing about 55,000 pounds. The figures are even more striking when we consider the cases of electric lighting and of the operation of alternators in parallel. For the former case, $\delta = \frac{1}{75}$, everything else being the same, the rim would weigh about 187,000 pounds, and in the latter, $\delta = \frac{1}{125}$, the weight would be not less than 330,000 pounds.

3. The manufacturing costs of the parts of large engines will be very high because special machinery is necessary. These in themselves are uneconomical because comparatively little used. To this must be added the uncertainty of successfully obtaining safe and sound machine parts of large dimen-

sions. Both of these things tend to increase manufacturing costs, besides making time of delivery uncertain.

4. Large castings, if at all complicated, give trouble owing to the formation of internal stresses. This disadvantage is augmented by unequal expansion by heat when in service. Unless carefully designed with full allowance for casting stresses and expansion, failure is likely to result, and has resulted in many instances. To these disadvantages must be added the difficulty of properly fitting such large pistons to prevent pressure losses and still have them loose enough to cause no great loss by friction, and the difficulty of properly oiling the piston surface.

5. The heat abstraction by the jacket water grows proportionately less as the cylinder volume increases. In order to prevent premature explosions, the compression pressure must be constantly reduced, which is accomplished, of course, by increasing the clearance. Now large engines nearly always use lean gas mixtures, in themselves difficult to ignite, and the increasing clearance spaces only add to the difficulty. Hence we find that engines generating from 50 to 100 H.P. in one cylinder show the best thermal efficiencies. Up to 150 H.P. there is no great increase in the efficiency, and beyond that there is a decrease rather than a gain.

The above considerations show clearly why above a certain rating more than one cylinder should be employed, this certain rating depending somewhat on other conditions. The main criterion seems to be the requirement regarding the co-efficient of regulations. For general power service one can go much higher with a single cylinder than for electric-lighting service, for instance, without seriously encountering any of the difficulties mentioned. For 4-cycle engines with any of the industrial gases the present ordinary limit seems to be about 300 H.P. in a single cylinder. For stationary engines of other types single cylinders may be employed up to from 50 to 60 H.P. The lower limit of division, however, depends somewhat on the individual circumstances of every case.

The methods of combining the cylinders are various. A

No.	Arrangement of cylinders and cranks.	No. of cylinders.	4-cycle. Angular movement of crank bet. explosions. Degrees.	4-cycle. Weight of fly wheel for Equal dimensions. x_1	4-cycle. Weight of fly wheel for Equal power x_2	4-cycle. I.H.P. of combination.	4-cycle. Weight of flywheel per I.H.P. Pounds.	2-cycle. Angular movement of crank bet. explosions. Degrees.	2-cycle. Weight of fly wheel for Equal dimensions (Sing. cyl. 4—cycle=1). x_3	2-cycle. Weight of fly wheel for Equal power (Sing. cyl. 4—cycle=1). x_4	2-cycle. I.H.P. of combination.	2-cycle. Weight of fly wheel, per I.H.P. Pounds.
I	Single cylinder, single-acting.....	1	720	1.000	1.000	65.8	147.0	360	.802	.401	131.6	59.6
II	Single cylinder, double-acting.....	1	180 & 540	1.230	.615	131.6	91.0	180	.424	.106	263.2	15.6
III	2 cylinder tandem, single-acting....	2	360	.769	.398	131.6	59.0	360	1.595	.399	263.2	59.2
IV	2 cylinder opposed, 1 crank, single-acting.....	2	180 & 540	1.290	.645	131.6	96.0	180	.335	.084	263.2	12.5
V	2 parallel cylinders, cranks together, single-acting	2	360	.792	.396	131.6	58.0	360	1.602	.401	263.2	54.6
VI	2 parallel cylinders, cranks 180° apart, single-acting.....	2	180 & 540	1.290	.645	131.6	96.0	180	.335	.084	263.2	12.5
VII	3 parallel cylinders, outside cranks together, inside 180° from other, single-acting.....	3	240	.678	.226	197.4	33.7	120	.237	.039	394.8	5.9
VIII	4 parallel cylinders, cranks 1 and 3 together, cranks 2 and 4 together, and 180° from 1 and 2, sing.-act'g..	4	180	.187	.048	263.2	7.1					
IX	4 cylinders, 2 pairs opposed, 2 cranks together, single-acting	4	180	.187	.187	263.2	7.1					

For a 2-cylinder opposed 2-crank engine, use values of Type V. Degrees between explosions = 360.

For the 4-cylinder, 2 pair opposed, 4-crank engine, use values of Type VIII or IX. Degrees between explosions = 180.

very good idea of the comparative merits of the different combinations may be gained from the preceding table, transposed from Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 328. The table is based on computations made on a Koerting 4-cycle engine using illuminating gas. The dimensions were 15 inches by 26 inches, 140 r.p.m. The weight of the fly wheel of this engine is taken as 1,000, and the table shows under

X_1 the relative weights of fly wheels of the various 4-cycle combinations when the cylinder dimensions are unchanged.

X_2 the same for 4-cycle combinations of equal horsepower.

X_3 the relative weights of fly wheels of 2-cycle combinations, as compared with the single cylinder 4-cycle, the cylinder dimensions being unchanged.

X_4 the same for 2-cycle combination of equal horsepower.

The coefficient of regulation δ is taken at $\frac{1}{40}$ in each case, and the velocity of the rim at 65.6 feet per second. The factors obtained therefore apply to an actual case, but they will serve with sufficient accuracy as a guide to the saving effected in fly-wheel weight in general cases.

The columns best compared are X_2 and X_4. They show, for instance, that if a certain horsepower is to be generated in one 4-cycle cylinder, and we call the weight of fly wheel required for $\delta = \frac{1}{40}$ equal to 1.0, we can do the same work in a double-acting 4-cycle with 61.5 per cent. of the fly-wheel weight, in a single-acting 4-cycle tandem with 39.8 per cent. of the fly-wheel weight, in a 3 cylinder single-acting 4-cycle of Type VII with but 22.6 per cent. of the original fly-wheel weight, and, if Type VII be made 2-cycle, but 3.9 per cent of the original weight is required.

In this connection Münzel in Stahl and Eisen, 1900, p. 528, gives a very interesting comparison of the first cost of *blast-furnace gas engines of 1,000-H.P., using various values for the coefficient of regulation, and various cylinder combinations.

*See also Güldner, p. 201.

RELATIVE FIRST COST OF 1000-H.P. BLAST-FURNACE GAS ENGINES.

Required regulation.	Single cylinder	2-cylinder parallel.	Deutz 2-cylinder opposed.	Deutz 4-cylinder opposed.
Ordinary operation, $\delta = \frac{1}{25}$	1	1.05	.90	.95
Electric lighting, $\delta = \frac{1}{75}$	1	.90	.85	.75
A-C operation, $\delta = 1\frac{1}{25}$	1	.75	.75	.60

Attempts have been made from time to time to improve the thermal efficiency of the gas engine either by obtaining complete expansion in one cylinder or by compounding. In either case the idea is to take advantage of the pressure still existing at the end of expansion in the cylinder in the ordinary case. The first method promises more gain than the second, but it can be shown in either case that the gain amounts to little or nothing. Complete expansion in the cylinder has been obtained by making the suction stroke shorter than the expansion stroke, *i. e.*, by closing the inlet ports before the suction stroke has been completed. This furnishes a very effective method of speed regulation, but fails of its real purpose, for at rated load the suction stroke is nearly as long as the expansion stroke and we revert to the original condition. At any lower load than the rated, the compression pressure is less and less as the load decreases, any gain in efficiency by more complete expansion will soon be negatived by a loss due to lower compression. The balance between gain on the one hand and loss on the other is reached sooner than indicated on account of other circumstances. The longer expansion makes the mean effective pressure less, correspondingly decreasing the specific power. This means larger engine dimensions for the same power, from which it follows that the friction losses are greater proportionately than they were before. To gain anything therefore the compression pressure should be kept constant and the complete expansion be made possible by other means than throttling the suction. The gain will be the greater the lower the original compression, the richer the charge, the greater

the degree of after burning, and the smaller the original friction losses in the machine.

The idea of compounding to gain complete expansion promises no gain whatever, although inventors have made attempts in this direction many times. The reasons for the failure are to be found in the high temperatures and low specific heats of the expanding gases. In order to obtain any reasonable mean effective pressure in the second cylinder, it is usually necessary to exhaust from the high-pressure cylinder at higher pressure than would ordinarily be done. The effect on the high-pressure cylinder is a higher mean temperature, and consequently greater proportionate loss to the jacket water, not to mention greater difficulty in proper lubrication. The very fact, however, that a greater initial pressure is thus obtained in the second cylinder has the great disadvantage that the gases flow over at a much higher temperature. In many cases the afterburning has not even been completed. In any event the valves and ports between the cylinders must be water cooled, if they are to last any length of time, and here is where we find the greatest loss. The abstraction of heat by the necessary water cooling, and the fluid friction in the ports causes so great a drop in pressure that the useful work in the second cylinder is very seriously reduced. These disadvantages seem to be inherent. As a consequence we find that compound gas engines already constructed have shown an efficiency about equal to that of a second-grade single-cylinder engine, and if any gain is to result from complete expansion, it is to be obtained rather in one cylinder than by compounding.

THE PRINCIPAL DIMENSIONS.

Having determined upon the nominal horsepower of the engine, and the fuel to be employed, it is at once approximately known how much of the fuel is necessary in a given time. This determines the amount of air required for combustion under assumed conditions, and in this manner the cubic contents of the charge in any given time, say one minute, becomes known. The generation of a certain horsepower therefore

narrows itself down to the determination of dimensions and speeds which will generate a certain cubic content in unit time. The variables involved under the control of the designer are the diameter of cylinder and the piston speed. The latter in turn resolves itself into stroke and revolutions per minute.

The above method of attacking the problem is much better than to lay out an ideal card, and by the use of card factors to determine the dimensions, as is done, and properly so, in the case of the steam engine. In the case of the internal-combustion engine just these card factors are extremely uncertain, depending upon composition of charge, perfection of ignition and of combustion, proper mixture of charge, etc. In actual practice it is found that these conditions will give a card factor varying from .4 to .8, and it is quite obvious that to weigh all the circumstances likely to attend each case, and from a multiplicity of allowances to determine a proper card factor, is little better than guessing. On the other hand, the first method requires only the determination of the approximate quantity of fuel and of air required for any determined power. Today the characteristics of the various fuels and of their combustion are so well known that no trouble is experienced, and the determination of engine dimensions from weight or volume of charge required would therefore seem to rest on the most natural basis.

In determining the dimensions and speed for any given charge, however, the designer is at once confronted by contradictory conclusions based upon thermodynamic and upon practical considerations. The former call for short stroke and high rotative speed, for under such conditions the time of combustion and of expansion is short, and the loss of heat to the jacket water will therefore be proportionately less. Practical considerations, however, call for long stroke and small diameter combined with the highest possible rotative speed. The reasons may be briefly cited as follows:

1. Short stroke and high speed excessively shorten the time of all events, which may prevent proper mixture of fuel and air, cause poor ignition, and reduce the capacity of the cylin-

der as a pump, if the 4-cycle be used, thus reducing the capacity of the machine.

2. The high compression pressures at present coming into use seriously reduce the cooling surface of the combustion chamber, since the inlet and outlet valve areas are not correspondingly reduced. A long stroke means a correspondingly long combustion chamber for the same end compression, thus furnishing greater cooling surface, and more room in which to place the valves.

3. The machine parts of an engine must be designed to withstand the maximum total piston pressure, while the capacity of the machine is determined by the total mean pressure. The economy of metal will therefore be the greater the nearer the mean pressure is to the maximum. The ratio $\frac{\text{maximum pressure}}{\text{mean pressure}}$ in the ordinary 4-cycle type is not at all favorable, varying in various cases from $\frac{14 - 18}{1}$, and for this type especially it becomes all important to generate the necessary volume for the required capacity with the smallest possible piston surface, in order to reduce the maximum total pressure, and thus to keep the machine parts as small as possible. Now the reduced piston surface requires an increase in either the stroke or the rotative speed in order to keep the piston displacement, *i. e.*, the capacity of the machine, the same. It is better to increase the stroke, because a higher rotative speed means a higher velocity of rubbing at the crank pin. This in turn means that, if heating is not to result, the rubbing surface must be increased in order to reduce the unit pressure on the pin. Lengthening the pin, however, leads to a greater diameter from considerations of strength. Higher rotative speeds therefore defeat the original design of keeping down the size of the machine parts.

All of this would lead to the conclusion on practical grounds that, contrary to thermal considerations, the ratio of $\frac{\text{stroke}}{\text{diameter}}$ should be as large as possible. The disadvantage of this is that a greater length of machine is required, but the advan-

tages gained in other places more than balance this, at least for stationary machines.

The following derivation of equations for valves of diameter, stroke and revolutions per minute is due to Güldner,* as is also the appended table. All metric measurements and values have been transformed to English units:

Let N_n = nominal brake horsepower.

n = r.p.m.

D = piston diameter in feet.

S = stroke in feet.

$V_h = .785\ D^2S$ = piston displacement per stroke in cubic feet.

$V = e_v\ V_h$ = actual volume of mixture in cubic feet per suction stroke, barometer 28.95 inches, temperature 59 degrees Fahrenheit.

$e_v = \frac{V}{V_h}$ = volumetric efficiency of suction stroke.

L = the volume of air in cubic feet required for 1 cubic foot of gas fuel, or 1 pound of liquid fuel, under most favorable practical conditions.

L_h = The resulting actual quantity of air in cubic feet for one combustion, at nominal horsepower.

C_s = the quantity of fuel used per hour, for gases in cubic feet, for liquids in pounds, for nominal horsepower.

C = the same per horsepower per hour.

C_h = the same per suction stroke.

H = the lower heating value of the fuel, for gases per 1 cubic foot, for liquids, per pound, in B.T.U.

$e = \frac{N_n \times 33{,}000 \times 60}{778\ C_sH} = \frac{2{,}545\ N_n}{C_sH}$ = thermal efficiency at the brake.

From the above we can derive for 4-cycle engines

$$C_s = \frac{N_n \times 33{,}000 \times 60}{778\ He} = \frac{2{,}545\ N_n}{He}; \quad . \quad . \quad (1)$$

*The article may be found either in die "Zeitschrift des Vereines deutscher Ingenieure," April 26, 1902, or in his book, Entwerfen und Berechnen der Verbrennungsmotoren, pp. 213–215.

$$C_h = \frac{2\,C_s}{60\,n} = \frac{2 \times 2{,}545\,N_n}{60 \cdot n\,H\,e} = \frac{84.8\,N_n}{n\,H\,e}; \quad . \quad . \quad (2)$$

$$L_h = \frac{C_s L}{30\,n} = \frac{2{,}545\,N_n L}{30\,n\,H\,e} = \frac{84.8\,N_n L}{n\,H\,e}. \quad . \quad . \quad . \quad (3)$$

For 2-cycle engines equations (2) and (3) should be divided by 2, because there is a charging stroke for every revolution.

ENGINES FOR GAS FUEL.

The charge taken in by the engine during one suction stroke will be

$$V = C_h + L_h.$$

This volume requires an effective piston displacement.

$$V_h = .785\,D^2 S = \frac{C_h + L_h}{e_v} = \frac{84.8\,N_n + 84.8\,N_n L}{n\,H\,e\,e_v}$$

$$= \frac{84.8\,N_n(1 + L)}{n\,H\,e\,e_v} \text{ cubic feet.} \quad . \quad (4)$$

Solving (4) for D, S and n in turn, we have

$$D = \sqrt{\frac{108\,N_n(1 + L)}{e\,n\,H\,S\,e_v}} \text{ feet}; \quad . \quad . \quad . \quad . \quad (5)$$

$$S = \frac{108\,N_n(1 + L)}{e\,n\,H\,D^2\,e_v} \text{ feet}; \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$n = \frac{108\,N_n(1 + L)}{e\,S\,H\,D^2\,e_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

ENGINES FOR LIQUID FUEL.

In this case the fuel is introduced either in liquid form or in the shape of vapor. In either case the volume ratios of fuel to air are very much smaller than they are in the case of gaseous fuels. Take for instance the case of alcohol, a fuel of low heating value. Here we find that the vapor does not form theoretically more than 4 per cent. of the volume of the

charge. In reality it is even less than this on account of the excess air used. In view of this fact we may put for these engines

$$V_h = .785\, D^2 S = \frac{L_h}{e_v} = \frac{84.8\, N_n L}{n H e e_v} \text{ cubic feet.} \quad . \quad (8)$$

Solving this again for D, S and n we obtain for engine using liquid fuel

$$D = \sqrt{\frac{108\, N_n L}{e\, n\, H S e_v}}, \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$S = \frac{108\, N_n L}{e\, n\, H D^2 e_v}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

and

$$n = \frac{108\, N_n L}{e\, S\, H D^2 e_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

In these equations the quantities to which certain values must be assigned are L, e and e_v. As stated before, there is now so much practical data at hand that the proper determination of these quantities should cause no difficulty. Table II* gives various values for e_v, the volumetric pump efficiency, and Table III* values of L as found for various fuels.

Since the nominal brake horsepower N_n is 18 to 20 per cent. below the maximum capacity of engines in ordinary cases, it is well to assume an excess of air of about 30 per cent. at the outset, in the case of machines using gas. In the case of liquid-fuel engines a greater excess, 50 to 60 per cent., should be used, because these fuels are usually high in heating value, and if any economical degree of compression is to be employed, the fuel mixture should be rather lean. Besides, it is not easy to produce a uniform fuel mixture in the case of liquid fuels, and an excess of air would help to overcome this difficulty. These excess air allowances have been made in the table. Table III also gives values of C and e for the various fuels and for various sizes of engines, as determined from existing data. The fuel consumed by heating apparatus

*Güldner, p. 175 and pp. 214 and 215.

and ignition flames is not included in C. In the case of engines using generator gas, however, the fuel used by the boiler furnishing steam for the generator has been taken into account, since a part of the heat is recovered from the steam in the generator.

TABLE II.

Type of engine.	Probable suction pressure, pounds.	Values of e_v.
Slow speed. Inlet valve mechanically operated	12.9—13.7	.88—.93
Slow speed. Automatic inlet valve	12.5—13.2	.80—.87
High speed. Inlet valve mechanically operated	11.7—12.5	.78—.85
High speed. Automatic inlet valve	11.4—12.2	.65—.75
Very high speed automobile engines, with automatic inlet valve, and air cooling	8.8—11.0	.50—.65

The ordinary vaporizer for liquid fuels will decrease the values of e_v from 3 to 5 per cent.

It will be of interest to see how the results obtained by use of these empirical formulae and tables agree with the dimensions of existing engines. For this purpose we will take some of the engines in the Mechanical Laboratory and a 140 H.P. producer gas engine tested by Mr. Buckingham in 1903. Two of the variables composing the necessary charge volume, *i. e.*, the stroke and the revolutions per minute, will be assumed, and the third, the diameter, is computed. It will be noted that the agreement is in all cases sufficiently close. It must be remembered, however, that as soon as the engine to be designed differs in any detail to any great extent, some of the assumptions must be modified to suit the altered conditions, and in this is where the experience of the designer finds its application.

TABLE III.

.....	Lower heating value (H) B.T.U. per		Air required, cubic feet. Theoretically. (L_0) per		Actually (L) per		Consumption of fuel (C) per brake horsepower hour and thermal brake efficiency (e), when N_n = 5 B.H.P.			10 B.H.P.			25 B.H.P.			50 B.H.P.			100 B.H.P. & over.		
							(C)		(e)	(C)		(e)	(C)		(e)	(C)		(e)	(C)		(e)
	cu. ft	*lbs.*	*cu. ft.*	*lbs*	*cu. ft.*	*lbs.*	*cu. ft.*	*lbs.*		*cu. ft*	*lbs.*		*cu ft.*	*lbs.*		*cu. ft.*	*lbs.*		*cu. ft.*	*lbs.*	
I. Illuminating gas.																					
a. Lean	505	...	5.5	...	7.5	...	24.8	...	.20	22.2	...	.22	20.4	...	.24	19.1	...	.26	18.5	...	.27
b. Ordinary	560	...	to	...	to	...	22.2	...	.20	20.1	...	.22	18.3	...	.24	16.9	...	.26	16.5	...	.27
	618	...		...		...	20.4	...	.20	18.3	...	.22	16.9	...	.24	15.5	...	.26	15.1	...	.27
c. Rich	675	...	6.5	...	10.0	...	18.7	...	.20	16.7	...	.22	15.5	...	.24	14.1	...	.26	13.7	...	.27
II. Generator gas.																					
a. Basis of anthracite	...	13,400	...	...	...	...	...	1.65	.11	...	1.43	.13	...	1.25	.15	...	1.10	.17	...	1.03	.18
b. Anthracite gas	141	...	.85	...	1.1	...	106.0	...	.17	95.3	...	.19	84.8	...	.21	77.6	...	.23	73.2	...	.24
c. On basis of coke	...	12,950	to	...	to	...	...	1.76	.11	...	1.49	.13	...	1.29	.15	...	1.14	.17	...	1.07	.18
d. Coke gas	129	...	1.00	...	1.4	...	116.0	...	.17	102.0	...	.19	91.8	...	.21	84.8	...	.23	81.3	...	.24
III. Blast-furnace gas	106	...	.75	...	1.0-1.2	...	..	...	...	130.0	...	.18	116.0	...	.20	102.0	...	.22	98.8	...	.24
IV. Coke-oven gas	505	...	5.3	...	7.0	...	...	...	...	35.3	..	.17	30.0	...	.19	26.4	...	.21	24.7	...	.23
V. Kerosene	...	18,900	...	185.0	...	257-353	...	1.25	.11	...	1.10	.12	...	.99	.13	...	...	...	...	...	...
VI. Crude oil (Diesel)	...	18,000	...	176	...	288-323	...	.55	.25	...	.527	.26	...	.506	.27	...	.462	.30	...	.440	.315
VII. Gasoline	...	19,800	...	176	...	240-323	...	.66	.19	...	.615	.21	...	.550	.23	...	...	...	...	...	...
VIII. Alcohol (90 per ct. vol)	...	10,300	...	96.5	...	128-193	...	1.10	.22	...	.99	.24	...	.92	.26	...	...	...	...	...	...

I. Hornsby–Ackroyd Kerosene Engine.

Brake horsepower = 3.
R.p.m. = 230, normal.
Stroke = 1 foot.

Assume $L = 300$ cubic feet, $e = 8$ per cent., as determined from tests, $H = 18{,}900$ B.T.U. per pound, and $e_v = .80$. The latter is placed at this low figure because the air enters through the heated exhaust ports.

$$D = \sqrt{\frac{108\, N_n L}{e n H S e_v}} = \sqrt{\frac{108 \times 3 \times 300}{.08 \times 230 \times 18{,}900 \times .80}}$$

$$= .55 \text{ feet} = 6.60 \text{ inches.}$$

Actual diameter = 7 inches.

II. Westinghouse Gas Engine.

2-Cylinder vertical.
Brake horsepower = 4 for each cylinder.
R.p.m. = 300.
Stroke = .66 feet.

Assume $L = 10$ cubic feet, $e = 22$ per cent., $H = 560$ B.T.U. per cubic foot and $e_v = .85$.

$$D = \sqrt{\frac{108 \times 4\,(1 + 10)}{.22 \times 300 \times 560 \times .66 \times .85}} = .471 \text{ feet}$$

$$= .471 \text{ ft.} = 5.65 \text{ inches.}$$

Actual diameter = 5.75 inches.

III. Acme Gas Engine.

Brake horsepower = 14.
R.p.m. = 220.
Stroke = 1.33 feet.

Values of L, e, H and e_v may be assumed, the same as for previous case.

$$D = \sqrt{\frac{108 \times 14\,(1 + 10)}{.22 \times 220 \times 560 \times 1.33 \times .85}} = .76 \text{ feet}$$

$$= 9.12 \text{ inches.}$$

Actual diameter = 9.00 inches.

IV. 140-H.P. Otto Gas Engine.

Brake horsepower = 140.
R.p.m. = 190.
Stroke = 3 feet.

Fuel is producer gas, 135 B.T.U. per cubic foot. Assume $L = 1.3$ cubic feet, $e = 23$ per cent. and $e_v = .85$ per cent.

$$D = \sqrt{\frac{108 \times 140\ (1 + 1.3)}{.23 \times 190 \times 135 \times 3 \times .85}} = 1.52 \text{ feet}$$

$$= 18.24 \text{ inches.}$$

Actual diameter = 19 inches.

IV. METHODS OF REGULATION.

The methods of speed regulation employed in internal-combustion engines may, for the purpose of discussion, be conveniently grouped under the following heads:

1. The hit-and-miss method.
2. Varying the composition of the charge to suit the load.
3. Changing the amount of charge to suit the load.

Other means are sometimes employed. In automobile machines especially speed is regulated sometimes by time of ignition of the charge, the ignition in some cases being completely suppressed. While this method of hand regulation is satisfactory for automobile and marine machines, it can not be said that the attempts to automatically control the time of ignition have been at all successful.

Two points should be kept in mind comparing the above methods of governing. The first of these is their economic efficiency, and the second the efficiency of regulation. Depending upon the service to which the engine is to be put, either one of these conditions may be of the higher importance, for it will in general be found that no one method combines highest economic efficiency with highest efficiency of regulation throughout the series of loads from no load to the rated.

1. The hit-and-miss method is mechanically one of the simplest. Usually a simple pendulum governor controls the

fuel valve, which is either opened or which stays closed during a revolution, depending upon the action of the governor. In some types the mixture-inlet valve is so controlled, and in others the exhaust valve is kept open during the "misses," meanwhile keeping fuel or mixture-inlet valve closed. Theoretically there should be at full load an explosion every 2 revolutions in the 4-cycle engine with this system of governing. At $\frac{3}{4}$ load the order should be: 3 explosions followed by 1 miss, at $\frac{1}{2}$ load 1 explosion followed by 1 miss, and so for all other loads in a regular manner. In general, if x be the fraction of the full load, y be the governing period, *i. e.*, the sum of hits and misses which follow each other other in rotation, and if z be the number of hits in this period, then we may put

$$x = \frac{z}{y}.$$

As long as the resistance encountered at the crank shaft is absolutely constant this law will be followed. But this is never the case, and as soon as the slightest disturbance occurs the governing period and the number of hits will be changed. At $\frac{3}{4}$ load for instance the sequence should be III—III—III—III—, where III represents the hits and — the misses. Any disturbance of the resistance above or below the average may result in the following arrangement III—II—IIII—II—IIII—. It can be shown that this at once causes a disturbance in the regulation.* It is of course impossible to control the accidental variations in the resistance to be overcome, and consequently this method of governing will always give somewhat unsatisfactory regulation. For ordinary power purposes this is, perhaps, not of great importance, especially when the prime mover is connected to the shafting or machinery by belting and when the driven machines themselves have masses of rotating metal. In such cases the elastic belts and the machines themselves tend to equalize the speed variations. For other work, however, as for electric lighting, this defect of the

*See Güldner, Verbrennungsmotoren, p. 317, and Mollier, "Zeitschrift des Vereines deutscher Ingenieure," p. 1704, Nov. 21, 1903.

hit-and-miss system is much more serious. It is of prime importance in such cases to carefully construct not only the governor, but also the mixing valves and the igniter, in order to make the number of misses between any two explosions as small as possible, and to be sure that the explosion following a miss occurs at the proper time with full force. But even under such conditions, it is not likely that any thing like satisfactory regulation for electric service, etc., can in a 4-cycle machine be attained by the hit-and-miss method, and Güldner advises to make the weight of the fly wheel rim twice as heavy as computations with allowable variation at full load require. This can, of course, not be commended from the standpoint of economy of first cost, and we find therefore that this method of governing is confined mostly to engines where the regulation need not be closer than $2\frac{1}{2}$ per cent. from full load to no load.

On the other hand, this method of governing is more economical of fuel than any other yet devised. The reason for this is apparent. If at full load the conditions, *i. e.*, amount of compression, composition of charge and point of ignition, are such as to give the highest efficiency for the fuel used, this efficiency should remain constant for each cycle from full load to no load, since none of the conditions, under which each cycle is described, are changed. Conditions extraneous to the working of the cycle of course change this, and we find that thermal efficiency decreases with the load. The fact remains, however, that, unless the conditions are abnormal, the efficiency is better sustained throughout the range of load in this system of governing than in any other. This is undoubtedly true for engines using gas or gasolene. In kerosene and alcohol engines it is quite probable that some of the fuel vapor is condensed on the comparatively cool cylinder surfaces after a series of misses at low loads, causing an increase in the fuel consumption, and a consequent decrease in efficiency.

2. Governing by varying the composition of the charge to suit the load is a method often employed, and, with proper fly-wheel weight, is found to give good satisfaction as regards

regulation. From the standpoint of fuel economy it is somewhat inferior to the hit-and-miss method. The same charge volume is used every cycle, the composition being changed by governor control of the fuel valve, the richer mixtures being used for the higher, and the leaner ones for the lower loads. The charge volume remaining constant, the amount of compression remains the same throughout, and the same thermal efficiency should therefore be realized from all cycles. As the load decreases, however, ignition of the leaner mixtures becomes increasingly difficult, and combustion is less complete and rapid. At the lower loads, cards from engines governing in this way sometimes show expansion lines which sink very slowly, showing an unusual amount of after burning.* Attempts have been made to avoid this by a simultaneous adjustment of the point of ignition, making it earlier, but to the writer's knowledge, this has not so far been very successful. We find, therefore, rather high fuel consumption at the lower loads in engines governing by this method, and the tendency today seems to be to desert this for other methods.

3. The third one of the principal methods of governing is to change the amount of the charge to suit the load, leaving its composition practically the same. This may be done in two ways. The mixture may be throttled throughout the suction stroke, in which case the pressure in the cylinder constantly decreases from beginning to end of suction. At full load with little or no throttling we shall have the regular card, as indicated in Fig. 1, for the 4-cycle machine. As the load decreases, the amount of throttling increases until we have a card as indicated in Fig. 2. The second way is to cut off the mixture completely some time before the suction stroke is complete, the time of cutting off depending on the load. In this case, too, we shall at full load have a card like Fig. 1, but at some lower load the card will be like Fig. 3, cut off occurring at b'. The second method is better than the first because less work is lost in the lower loop. They both possess the disadvantage that the compression varies with the

*Meyer, "Zeitschrift des Vereines deutscher Ingenieure," p. 602-603, April 25, 1903.

load, decreasing as the load decreases. Consequently, everything else being the same, as the load decreases the engine works with cycles whose efficiency constantly decreases. This defect in this method of governing is to some extent compensated for by the fact that more complete expansion is obtained as the load decreases, and it may be said that, considering both efficiency of regulation and economy of fuel, this method is the best of the three. For closeness of regulation this is undoubtedly true. The question as to how much inferior this method is to the hit-and-miss system as regards fuel economy led Meyer to make the following interesting comparison between the two, in connection with his Report on Tests of Alcohol-locomobiles for the German Agricultural Society, 1902.* Tests were made on a Deutz engine using the cut-off system of regulation, and the following figures, transposed to English units where necessary, were obtained.

Load.	R.p.m.	Brake horsepower. (Metric.)	Per cent. of full load.	Vacuum at end of suction stroke.	Pressure at end of compression above atmosphere.	Per cent. cut-off = $\frac{a'd''}{a''i'}$ Fig. 3.	Mean effective pressure of the area $a'b'c'$ $d'i'h'a'$ Fig. 3.	Work lost in per cent. of net area of diagram.
				Pounds.	*Pounds.*		*Pounds.*	
Full	278	15.95	100	2.40	201.0	93	2.70	.10
¾	284	12.18	74.5	5.25	134.5	71	2.55	1.80
	286	10.26	62.4	5.80	124.5	66	2.55	2.90
½	286.5	8.18	49.8	6.80	106.0	56	2.70	5.40
	289.5	6.19	37.3	9.06	88.0	42	2.55	12.10
¼	294	4.22	25.0	10.05	56.8	33	2.42	20.30
	296	2.07	12.2	10.75	54.0	24	2.13	34.3
Friction		0	0	11.50	31.2	19	1.84	48.4

Meyer makes the comparison by determining how much work is lost when a given amount of mixture is in the one case worked through cycle Fig. 1, *i. e.*, the hit-and-miss system, and in the second case through cycle Fig. 3, *i. e.*, the cut-off system. The areas of the cards measure the work de-

*Meyer, "Zeitschrift des Vereines deutscher Ingenieure," p. 602–603, April 25, 1903.

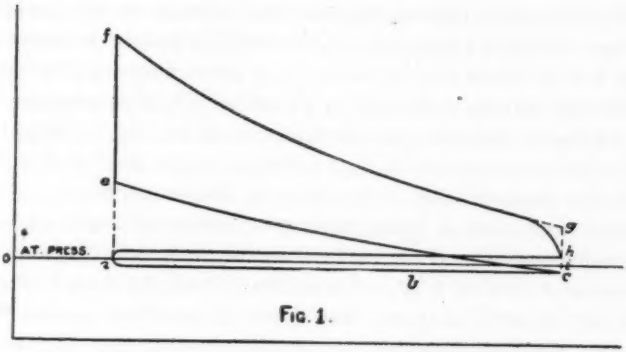


Fig. 1.

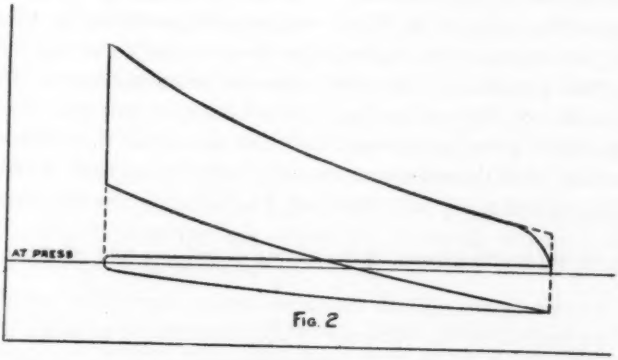


Fig. 2

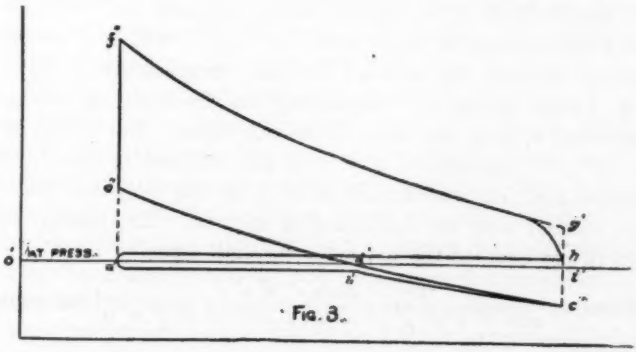


Fig. 3.

is thermodynamically desirable. The lean composition of the mixture makes any danger from pre-ignition remote. Being a thermodynamically more correct system of governing, better results might have been expected, but tests have shown that while the results attained are very good, they are no better than those attained by first-class engines using other systems of speed regulation. (See Güldner, Verbrennungsmotoren, p. 424.)

V. ALCOHOL AS A FUEL.

For the American power user the question of the utilization of alcohol in internal-combustion engines has at present very little commercial interest, and this state of things is likely to continue for a long time. For many European countries, more particularly Germany, the conditions are, however, very different. There crude oil, kerosene and benzine, owing to duty, are nearly on a level in price with alcohol, which is there a very important domestic product. As will be seen below, alcohol can be bought in moderately large quantities even cheaper than the other fuels mentioned. Germany also exhibits the rather queer spectacle that the engine builder did not come to use alcohol because he recognized in it a good fuel for his engine, but because the distiller, owing to overproduction, had to find use for his product. This condition of things is not natural, and there is, consequently, some question as to its stability.*

However that may be, and whatever the present-day interest of the American power consumer, the manner in which the problem of the utilization of alcohol was solved, the results attained and the present status of the question should be of interest to the American engineer.

Since 1887 several things concurred to cause an annual overproduction of alcohol in Germany. The potato crop, from which in the Eastern provinces most of the alcohol is made by agricultural proprietors, increased about 33 per cent. during the last ten years. An increase in the revenue tax on

* R. Schöttler, "Zeitschrift des Vereines deutscher Ingenieure," p. 1157, Aug. 2, 1902.

alcoholic beverages lowered the domestic consumption of alcohol, and an increase of import duty imposed by foreign countries caused a decrease in the exports. The potato crop, after the demand for them as food has been satisfied, is of little value except for the making of alcohol; under these circumstances the supply rose while the demand decreased. The natural outcome was to seek new fields for the utilization of the product, and in consequence the amount of alcohol used for technical purposes increased from about 10,000,000 gallons in 1887 to 30,000,000 gallons in 1901.

The price naturally decreased during this period, but it fluctuated so much that no decided step formed in the utilization of the product as fuel for gas engines was made until about 1899. Today, under a schedule of prices fixed by what appears to be a company controlling the alcohol supply, 90 per cent. alcohol can be bought in the consumer's own vessels, and in lots of not less than 1,600 gallons, for $3.51 per 26.4 gallons, which is about 13.3 cents per gallon. This price prevails from November 1 to May 15; for the remainder of the year the price is $3.74 for the same quantities, corresponding to 14.2 cents per gallon. It is guaranteed that these prices will be maintained until 1908. Under such conditions alcohol can compete with benzine, gasolene or kerosene, at the prices prevailing in Germany for the latter fuels. As Schöttler remarked, however, and as some engineers of authority, Diesel, for instance, strongly believe, the condition of things is unstable, since nobody knows what will occur after 1908. Diesel* seems to believe that alcohol will never be a serious competitor of either benzine, gasolene, kerosene, or crude oil, even in Germany.

In comparing the various liquid fuels above mentioned, regard should be paid to the following points:

1. Cost of the various fuels on the basis of heat units, say 10,000 B.T.U.

2. Cost per delivered horsepower per hour.

*"Z. d. V. deutscher Ingenieure," p. 1367, Sept. 19, 1903.

1. COST OF THE VARIOUS FUELS ON THE BASIS OF HEAT UNITS.

Favre and Silberman determined the lower heating value of absolute alcohol to be 11,664 B.T.U. per pound. Commercial alcohol is, however, never absolute, containing various quantities of water. Besides this, the German law requires that alcohol used for technical purposes in order to be free from revenue tax shall be rendered undrinkable by the addition of about 2 per cent. wood alcohol and ½ per cent. of Pyridine bases. These admixtures change the heating value. Taking into account the water only, it lowers the heating value in two ways, first by replacing some alcohol, and secondly, by requiring heat for its own vaporization. The following table, due to Schöttler, gives the variation in the heating value and in the cost of alcohol of varying purity. The price is taken at about $3.74 per 26.4 gallons for the 90 per cent. alcohol.

Absolute alcohol, volume, per cent.	Spec. gravity.	Absolute alcohol weight, per cent.	Lower heating value. B.T.U. per pound.	Cost, cents per gallon.	Cost, cents per pound.	Cost of 10,000 B.T.U., cents.
95	.805	93.8	10,880	16.3	2.44	2.24
90	.815	87.7	10,080	15.4	2.28	2.25
85	.826	81.8	9,360	14.5	2.12	2.26
80	.836	76.1	8,630	13.6	1.96	2.27
75	.846	70.5	7,920	12.8	1.82	2.29
70	.856	65.0	7,200	12.0	1.69	2.35

Comparing the above figures with the cost of gasolene and kerosene, Meyer gives the price of gasolene as 24 Pf. per kg., which is 2.64 cents per pound, or about 15.5 cents per gallon, the cost of kerosene is given as 22 Pfg. per kg., which reduces to 2.43 cents per pound, or about 16.2 cents per gallon. The heating value of these two fuels is about the same, and may be taken at about 18,500 B.T.U. per pound on the average. This will make the cost of 10,000 B.T.U. for gasolene 1.43 cents, and for kerosene 1.31 cents. Diesel gives the price of paraffin oil, or crude oil, for Germany at from 8.25 to 10 Pfg. per kg., which corresponds to from .88 to 1.06 cents per

pound. He refers here to domestic oils, the cost of the imported product being higher on account of import duty. Assuming 18,000 B.T.U. as the average heating value of the crude oil, the cost of 10,000 B.T.U. will be from .48 to .60 cents.

The following table gives a summary of all the figures. For the purpose of comparison the cost figures for the U. S. have been appended. For these columns, the prices were taken as follows :

Gasolene 13.5 cents per gallon in barrels of from 50 to 56 gallons.

Kerosene 12.5 cents per gallon in barrels of from 50 to 56 gallons.

Crude oil, $2.50 per barrel of 50 gallons at New York.

The figures for alcohol are not given, for with ethyl alcohol at $2.50 per gallon, and methyl alcohol at $1.25, there can be no rational basis for comparison.

.....	Cost per gallon, cents.		Cost per pound, cents.		Cost of 10,000 B.T.U., cents.	
	Germany.	U. S.	Germany.	U. S.	Germany.	U. S.
Gasolene.....	15.5	13.5	2.64	2.32	1.43	1.25
Kerosene.....	16.2	12.5	2.43	1.88	1.31	1.02
Crude oil, sp. gr. = .9..	8.0	5.0	1.06	.66	.60	.36
Alcohol, 90 per c. vol...	14.2	...	2.28	...	2.25	...

2. COST PER DELIVERED HORSEPOWER PER HOUR.

The highest economy so far shown for alcohol was obtained on the tests of the Deutz alcohol motor by Prof. E. Meyer. For purposes of comparison he added to these figures the highest economy obtained with gasolene and kerosene engines. Diesel quotes this table in an article in the "Zeitschrift des Vereins deutscher Ingenieure," Sept. 19, 1903, and adds to it the best economy figures obtained by the Diesel engine on crude oil. The annexed table shows the results transfer-

red to English units. Diesel remarks that on later tests the consumption of his engines has been decreased about 15 per cent. under that given in the table.

		Other cycle engines.			Diesel Engine.
		Gasolene.	Kerosene.	Alcohol.	Paraffin oil.
Cost of fuel per lb., cents.....		2.64	2.43	2.28	1.06
Cost of 10,000 B.T.U., cents.		1.43	1.31	2.25	.60
Ratio, crude oil—1.....		2.40	2.20	3.75	1.
Best..... Consumption..... lbs. per H.P. hour.....	Full L'd..	.654 (Banki, .488)	.727	.805	.450
	Half L'd..	.955	1.082	1.117	.533
Increase in consumption, full load to half load, p. c..		46.	49.	44.	19.
Cost per..... Delivered..... .. Horsepower per hour, cents.....	Full L'd..	1.73	1.77	1.83	.477
	Half L'd	2.52	2.64	2.54	.565
	Average..	2.12	2.20	2.19	.521
Ratio, crude oil = 1.....		4.06	4.21	4.20	1.
Hig'st thermal efficiency per D.H.P. so far obtained.....	Full L'd,p.c	20.5	17.6	32.7	32.6
	H'f L'd, p.c	14.0	11.8	22.7	27.4

The table shows in the first place that crude oil, as burned in the Diesel engine, is the cheapest of the fuels, the ratio being about 1 to 4 in its favor. This superiority probably holds good also when this oil is burned in an explosion engine, in contradistinction to the Diesel, provided that such an engine can successfully handle this fuel. In a test made on an engine using the Otto cycle by the writer in New York during the summer of 1903, for instance, the consumption of oil for full load was .488 pounds per B.H.P. per hour, and for half load, about .735 pounds. This, at the rate of 5 cents per gallon for the oil at New York, corresponds to a cost of .321 cents per B.H.P. at full load, and .485 cents at half load, an average of .403 cents. Crude oil would therefore seem to be the fuel of greatest promise, but there are a great many practical difficulties in the way of its successful application, although it is claimed that it is burned successfully in a number of types of engines.

The table also shows that in spite of the fact that 90 per cent. alcohol costs about 60 per cent. more per 10,000 B.T.U.

than either gasolene or kerosene, as regards the cost per delivered horsepower, *when each is burned in an engine designed for its use*, they are all on the same level. The reason for this is not far to seek. The thermal efficiency of the Deutz engine was 32.7 per cent., as compared with 20.5 per cent. for the gasolene and 17.6 per cent. for the kerosene engine, that is, a properly designed alcohol engine will utilize 50 per cent. more of the heat furnished it than a gasoline engine, and nearly 100 per cent. more than this last engine when using kerosene. In explaining this result, Meyer determined first that the work lost in fluid friction, friction of the machine, etc., was about the same in the alcohol engine as it was in the one which was used for the gasolene or kerosene tests. He further showed, that since the degree of compression was the same for both the kerosene and gasolene tests, the greater consumption of the kerosene can only be explained on the assumption that some of the kerosene goes through the engine unburned, the vapor being condensed as it enters the cylinder. This being so, the greater thermal efficiency of the alcohol engine can only be due to differences in the cycle, and here we find that a much higher degree of compression can be employed with alcohol than with gasolene or kerosene. This is due to the fact that the mixture of alcohol and air possesses a rather high specific heat, the percentage of water contained in the alcohol further increases this, and in consequence a greater degree of compression can be used without the fear of premature explosions. While the ratio of compression for gasolene or kerosene can not be carried much above 4, that for alcohol may with ease be made 10, and this in itself would mean on theoretical grounds an increase in the efficiency equal to about 50 per cent. As is well known, Banki increases the ratio of compression for gasolene by injecting water vapor, and has in that way reached a degree of compression equal to 9.83, the consumption of gasolene being .488 pounds per B.H.P. per hour, corresponding to a cost of 1.29 cents per B.H.P. per hour. Against such results alcohol will, of course, never be able to compete, but aside from this it is in Germany practically the equal of gasolene and kerosene as regards economy.

For automobile purposes, however, the Banki principle can not be used on account of the large quantities of water which must be carried, and here, as also for general power purposes, alcohol possesses several advantages over the other fuels. It is practically odorless, and much less inflammable than gasolene. As compared with korosene, it is much cleaner, there being no carbon deposit formed. Air even at ordinary temperatures takes up considerable alcohol vapor, so that the danger of condensation in the cylinder is minimized. There is, of course, danger of rusting through the use of alcohol, but proper care should render this disadvantage of little moment. Rusting may also be prevented by the addition of benzol (C_6H_6), and tests have shown that the cost of operation with a mixture of 15 per cent. by weight of benzol and 85 per cent. of alcohol is a little lower than the cost when alcohol alone is used. It has, however, been noticed that engine needs cleaning more often when such a mixture is used. A further disadvantage in the use of alcohol lies in the fact that it must be vaporized through heat, and it therefore becomes necessary either to have an open flame to preheat the vaporizer, or to start with gasolene. The latter method is preferred to the former, there being less danger, in the case of traction engines or other agricultural machinery, and also because the change from gasolene to alcohol can ordinarily be made by throwing over a single lever.

While Meyer's results thus point to the practical equality of alcohol as a fuel for gas engines, he himself has pointed out that under the high degrees of compression employed, the engines are very sensitive as regards the proper amount of fuel taken per stroke, the explosions easily becoming excessive, and he asks whether it would not be better to use somewhat lower compression to insure greater certainty of operation, of course at the cost of economy. As a matter of fact, comparative tests made in Nürnberg* on a 2-H.P. gasolene engine in which the compression could be changed from 50 pounds to somewhat

*"Z. d. v. deutscher Ingenieure," p. 59, Jan. 10, 1903.

above 105 pounds have led to different results. The conclusions from these tests are briefly as follows:

1. The consumption of alcohol is even at the high compressions greater than the consumption of gasolene.

2. In order to approach the operating cost of a gasolene motor, the alcohol motor requires a much higher compression. This has the disadvantage that a much heavier engine is needed to stand the work.

3. The mixture of alcohol vapor and air is much more difficult to ignite than the gasolene vapor. It is therefore necessary to employ gasolene on starting.

4. The tests have not shown that less fouling of the engine takes place with alcohol than with gasolene. On the contrary, a great deal of water was found in the ignition canal and exhaust passages after long operation with alcohol. Analysis of the exhaust gases also showed small quantities of acids, as acetic acid, whose action on the iron parts can be but unfavorable.

Of these conclusions, the second is well taken, and has its greatest bearing in the case of automobile machines; the third has already been mentioned. The first can perhaps be explained on the ground that the engine was much smaller than that used by Meyer; the fourth, however, is in direct contradiction to the experience of other experimenters. On the basis of these experiments the final conclusion is reached "that equality of operating cost as between alcohol and gasolene is reached only when the compression pressure in the alcohol motor is at least 120 pounds per square inch, and even then 20 per cent. of benzol must be added. The alcohol engine can not seriously compete with the gasolene engine, especially for automobile purposes and other small units."

Diesel also attacks the use of alcohol, and on double ground. Referring to his engine, which he claims can burn any of the liquid fuels to equal advantage, he states that price of alcohol would have to decrease in the ratio of 4 to 1 in order to compete with the Diesel engine. This can not, of course, be expected; but by the same process of reasoning the

same conclusion also applies to gasolene and kerosene. The next statement, that even if Germany produced no paraffin oils, the operation of the Diesel engine with the taxed imported kerosene would cost only half as much as the operation of an alcoholic engine, carries more weight. He states also that the present price of alcohol in Germany is artificial and causes a loss to the producer, the loss being evened up by the higher cost of alcohol not used for technical purposes, *i. e.*, at the cost of third parties, and concludes by pointing out that, in nearly every other country, with a few exceptions, the cost of the liquid fuels is so much less than in Germany, that alcohol would have to cost very much less than it does now in order to enter into competition. Taking crude oil at 5 cents per gallon at New York, for instance, alcohol would have to cost about 2.5 cents per gallon in order to cost the same on the basis of heat units.

PRACTICAL AIDS FOR THE NAVAL ENGINEER.

By Lieutenant H. T. Winston, U. S. N., Member.

All officers who have been assigned to engineering duties in the Navy, whether they be young, and have had little previous engineering experience, or old and experienced engineers, particularly if they have been on shore duty for some time, must realize that there is something lacking in the available text books, articles on engineering, and the engineering instructions of the Navy Regulations. These text books, though excellent in their own line, and which a good engineer must be familiar with in order to understand his machinery and its care, are intended primarily for midshipmen, and they fail to give in concise form a practical scheme for the proper internal workings, the coordination of different subdivisions, and for the day-to-day work of the engineer's division.

It will be the purpose of this article to give such a scheme, and in addition a few sketches and tables, all of which have been tried out by the writer and found to be of much benefit, in so far as smooth working of the engineer's department is concerned. In addition, it has been found that the men become much more proficient, a heavy tax is removed from the engineer's memory and he has considerable more time for study, for other important duties, or for leisure if he so elects.

While in some respects what follows applies only to a large vessel, and particularly to the *Charleston*, with her four-watch system, twin screws, four firerooms and four 4-furnace B. & W. boilers in each fireroom, still it is believed that this article is for the most part of general application, and its purpose will have been accomplished if suggestions of value

have been made to engineer officers, though they may change the scheme to suit their own ideas and to meet varying conditions.

One of the most successful and largest contracting firms in New York, which erects sky-scrapers and contracts for similar work, gives to each employee in its head offices a typewritten copy of the rules which are to guide him in his work. A preface to each set of rules states that their issue has been made necessary by the fact that, whenever anything has gone wrong in the office, an investigation showed some of these rules had not been observed, and that it was desired to avoid in the future the consequent "unpleasantness" for the employee. As in private business, so in the Navy, written instructions serve to avoid "consequent unpleasantnesses," particularly as the men are often green and have had little opportunity to acquire knowledge of routine work, except through long experience and hard knocks.

ENGINE-ROOM AIDS. (See Appendix A.)

The officer of the deck of the engineer's department is usually at the desk in the starboard engine room, and near by should be found the following aids:

Port and Sea Routines (preferably in a bulletin board).

Book of Routine Orders (hanging over desk).

Morning Order Book (contains for the night and following morning orders additional to the routine and routine orders).

Muster Lists of Steaming Sections (in the desk).

Tables of Water in Double Bottoms, per inch, as sounded (in the desk).

Booklet of Skeleton Pipe Plans (in the desk).

Table showing Heaters and Pantries on each Heating Circuit (hanging up).

Diagram of Drainage System and Accompanying Table (in a bulletin board).

A Clinometer so situated that it is easily seen.

On a board (about 18 inches by 12 inches) post auxiliary

watch list, lists of special working parties or details, sick, confined, absentees, cooks, and special notices.

NOTE.—As the notices for this last board are frequently changed, it will be found that a piece of ½-inch oak or pine, studded with nails driven through from the back so the points project about ¼ inch and are 2 inches apart, makes a very convenient kind of board.

DUTIES OF YEOMAN AND SPECIAL DETAILS.

In appendix "B" are given rules for the engineer's office, storerooms, etc. It is suggested that the different sets of rules be posted where the special details and others stand their watches or perform their duties. By doing this there will be a similarity in the performance of duty by the different men, and any new man on a station will have before him general rules for his guidance and will be unable to plead as an excuse ignorance of previous orders.

The "Duties of the Yeoman and his Assistants" is deserving of special attention on the part of the engineer officer, as it contains many things the senior engineer must remember. These items have been purposely omitted from "Aids for the Engineer's State Room" in order to avoid duplication.

Perhaps some officers will think that the keeping of a "Coal-bunker expenditure book" is a useless amount of trouble, but I have used this system on two ships (*Charleston* and *Newport*) and found that it was about the only satisfactory way in which to keep the coal account straight. After a few days the yeoman becomes accustomed to this method, and work is really saved instead of added, because the books show the actual amount on hand. It might be noted further that with a separate coal account for each fireroom, mismanagement or poor firing in that fireroom or an incorrect coal tally shows up at once. As a parting shot, I cannot help adding that tally boards for the number of buckets of coal should, of course, be used in each fireroom.

ENGINEER'S BULLETIN BOARDS.—(OUTSIDE ENGINE ROOM.)

The subject of watch, quarters and station bills will not be taken up in this article, with the exception of a few suggestions, since they must vary considerably on different vessels, and this matter has already been treated by writers in the JOURNAL and by Commander Barton in "Naval Engines and Machinery." The suggestions follow:

Make the bills as simple as possible; let the "steaming sections" be the framework of the whole structure, with the men at their steaming stations for all duties as far as possible; in changing from steaming to auxiliary watches do not change the men on watch, but simply reduce the number (and *vice versa*); have the liberty lists so made up that men will go ashore when off watch duty and will stay on board when on watch duty.

A very convenient form for the bills is one which allows the easy shifting of men's stations. This may be provided for by having each man's name on a separate piece of card board held in place by brass clips so the names may be shifted. By making these clips long enough and having other pieces of card board on which are printed "sick," "absent," "confined," etc., it may be noted at once whether a man is available for duty or not.

It is hardly necessary to add that all bulletin boards should be located where they are conspicuous and may be easily consulted by all members of the engineer's force. Some of them also will require frequent visits of the yoeman, and should be near the engineer's office.

One or two special bulletin boards will be found necessary for the posting of notices, orders affecting the whole division and miscellaneous information; for example, "sea and port routines," "liberty lists," "special requests for liberty to be made before 11·00 A. M.," "steaming shoes to be taken off before going on deck," etc.

AIDS FOR SENIOR ENGINEER'S STATE ROOM.

Appendix "C" contains matter which will be found useful to the senior engineer personally, and may be kept in his room for ready reference.

The *routines* show what he must attend to apart from what has been stated heretofore, and may also be used as check lists, the different items being checked off as the work is done and the dates noted if so desired.

The *coal and water consumption* at various revolutions is often a vexatious problem, and the table given will be found useful in solving the problem. (The writer uses curves from which this table has been taken.) With the data from the progressive or measured-mile trials of the ship it is a simple matter to construct the necessary curves and with them to tell approximately how much coal and water will be used at a given speed, except at very high or low speeds. Also taking the amount actually burned in a day it may be seen at a glance whether one has an economical performance or not.

For this ship (*Charleston*) it has been assumed that ordinarily the auxiliaries (apart from the main engine and its necessary auxiliaries) require at sea an average of ten tons of coal per day, which is perhaps a rather low estimate, since the coal consumption in port averages about fifteen tons. The small amount of accurate data available shows that the actual coal consumption at sea runs, under ordinary conditions, slightly above and almost parallel to the 2 pounds per I.H.P. curve. In this connection, it may be interesting to note that the steam consumption of the *Charleston's* port engine was increased about 30 per cent. by a badly scored H.P. cylinder liner and scored liner of H.P. valve chest. A calculation of the steam consumption from indicator cards taken from both engines when running at 90 turns (nearly one-fourth full power) gave the above result, and steam gauges showed that the port H.P. valve leaked so badly that the drop in steam pressure in H.P. cylinder was only about 8 pounds.

"*Books and Notes for Reference*" gives a list of what the engineer will find desirable and useful in looking up various subjects.

Estimating coal in the bunkers is at the best an approximation, and some "landmarks" in the bunkers, particularly if they have an irregular cross-section, will aid in making the estimate fairly accurate. The sketch shows a sample of such "landmarks," the capacities up to them being calculated from measurements taken from the ship's drawings, and assuming that coal can be stowed to within nine inches of the deck overhead.

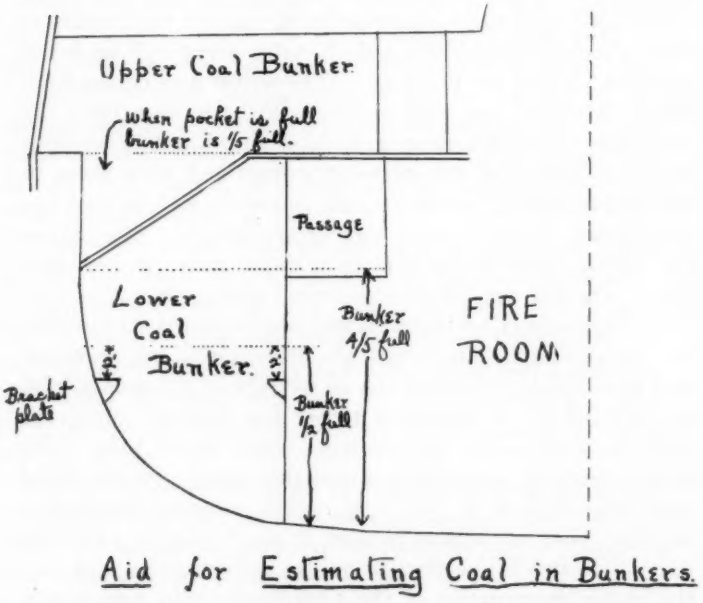


Fig. 1.

Setting the links to obtain proper distribution of power is another proposition which must be tackled by the engineer To obtain the most economical performance the H.P. links will be run in as far as possible and the throttle kept practically wide open. While the writer has not experimented sufficiently in this line to assert that the curves shown are entirely correct, he has found them of considerable assistance on the

Charleston, where the speed has not varied greatly during a cruise around South America. It is believed that by running with the throttle wide open, adjusting the H.P. links to give two or three different speeds, and setting the other links to give an even distribution of power, fairly accurate curves could be plotted from the points obtained.

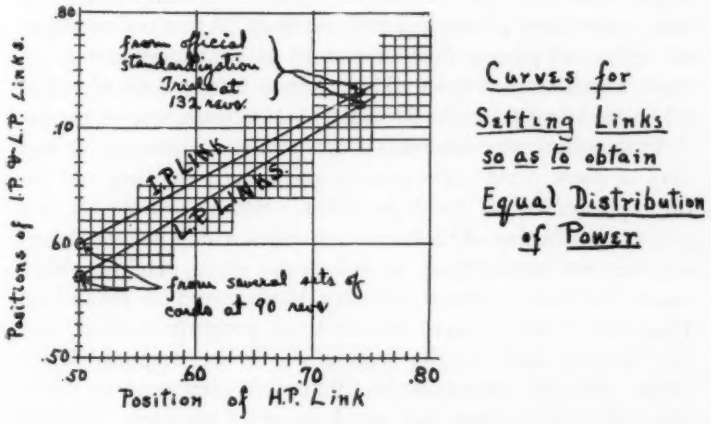


Fig. 2.

The straight lines have checked up closely for a few different speeds with the *Charleston*. Of course, it should be remembered that the pressure in the H.P. valve chest must be constant, and that different results will be obtained with steam on and off the jackets. The rule given by Lieutenant H. C. Dinger, U. S. N., in his article in Vol. XVIII, No. 4, p. 144, of the JOURNAL, with regard to estimating the horsepowers of the different cylinders of the engine, holds only rather roughly when there is much drop from the L.P. valve chest to the condenser and from the I.P. to L.P. valve chests. Such is the case on the *Charleston*, and the horsepower of L.P. cylinders is largely overestimated from the gauge readings.

MISCELLANEOUS NOTES.

There remain in addition to the above many other questions which may be perplexing, particularly to the line officer doing engineering duties for the first time. Among these are what duty will be required of certain special mechanics. For example, the coppersmith should understand that he is particularly responsible for all copper piping (especially salt-water lines), that he is to keep properly repaired all oil cans, oil trays, oil lights, oil piping, dust protectors of auxiliarymachine ry, etc. The blacksmith should keep in a proper state of repair all boiler-firing tools, lathe tools, coal shovels, wrenches, etc.

Throughout the engineer's department numerous things may be done to aid in the smooth and orderly running and the tidy appearance of the department. Some of these are the proper distribution of tools and wrenches which belong to certain stations or machines, as evaporator room, machine shop, steam launches, fireroom, fireroom blowers and ice machines. Each station should have its own tools, properly marked, and the leading man on the station should be responsible for them. Should anyone desire other tools they must be taken from the store rooms and not from other stations. Locked tool chests in each fireroom and on other stations will be found most useful in preventing the loss or mislaying of tools.

Small lockers, even if they are little more than pigeon holes, for steaming shoes in the engine-room hatch are a great convenience for the men and prevent trouble about dirtying up decks.

Good canvas screens for the crankpits, with suitable stops, heavily painted on one side with asphaltum and on the other with brown zinc, tidy up the engine room in addition to keeping the floor plates dry.

Suitable permanent clothes lines, located where the clothes will not be soiled or water drip on machinery, are easily rigged They will be found a convenience to the men, and with them clothing will not be scattered around the Department.

Considerable attention may be paid to auxiliaries both in and outside the Department. Some of them use an excessive

amount of steam even when well cared for (see Commander W. W. White's article on tests on board the *Minneapolis*). With improper care this consumption of steam (or coal) is easily doubled or trebled. Another point in connection with auxiliaries may be illustrated by an incident on board the *Charleston*. It had been found that considerable oil came from the dynamo engines to the feed tanks, but the cause was difficult to discover. Finally, one day an electrician was caught emptying a bucket of oil and water that he had taken from inside the engine casing by simply opening the drains to the L.P. cylinder of the engine and letting the vaccum draw it into the condenser. It is hardly necessary to add that this act of supreme laziness and gross ignorance was discontinued.

APPENDIX A.

PORT ROUTINE.

5·30 to 7·30 A. M.	8·15 to 11·30 A. M.	1·15 to 4·30 P. M.
Monday. Men who wish to, may scrub clothes until 6·00 A. M. Hoist ashes. Pump all bilges dry, clean stations, and work of day as ordered.	Move main engines and auxiliaries not in use. Work of the day. Test water in boilers in use and in feed tanks. Give boilers in use surface and bottom blow and test try cocks. Clean grease extractors. Blacksmith overhaul firing tools.	Finish uncompleted routine work and work of the day.
Tuesday. Same.....	Move main engines and auxiliaries not in use. Work of day. Move all cocks and valves not ordinarily moved. Overhaul steam traps.	Same.
Wednesday. Same.....	Move main engines and auxiliaries not in use. Overhaul W. T. doors and mechanical devices. Test deck operating gear of W. T. doors and of boiler stop and safety valves. Give boilers in use surface and bottom blow and test try cocks.	Half holiday if there is no urgent work.
Thursday. Same.....	Move main engines and auxiliaries not in use. Estimate coal on hand and inspect bunkers, scaling and painting when necessary. Work of day. Clean grease extractors.	Uncompleted routine work and work of the day.
Friday. Hoist ashes. Clean bilges and bilge strainers. Blacksmith overhaul machine shop tools and wrenches of Dept. Coppersmith inspect salt-water piping and repair oil cans, oil lights, etc.	Clean bilges, inspect D. B. compartments and renew gaskets of manholes when necessary. Test main and auxiliary drains by pumping out bilges, and test gear for operating main drain valves from deck above.	Same.
Saturday. Hoist ashes. Field day. Scrub paint work.	Move main engines and auxiliaries not in use. Scrub paint work and clean stations.	Half holiday.
Sunday. Hoist ashes, pump out bilges and clean stations.	Before quarters clean stations and prepare for Sunday inspection.	

NOTE.—Whenever a job of work is completed or work is discontinued for the day, all tools and stores must be returned to the storerooms and wrenches to wrench boards.

Every Day.—Before 7·50 P. M.—Storeroom keepers report unuseds tores, and tools returned to storerooms. Close all W. T. doors and manholes not to be used during the night. Take down all clothes hanging in unauthorized places. Sound D. bottoms and log fresh water on hand. *At 7·50 P. M.* report department secure.

Night Watches.—8·00 to 12·00 P. M.—Watch cleans all number 1 and 3 furnaces. *Mid. watch* sweeps tubes of boilers in use, except Saturday night. Morning watch cleans all number 2 and 4 fires.

SEA ROUTINE.

5·30 A. M.—Call working party, detail for trimming down coal, machine-shop men, cooks, washroom keepers and other men who do not stand watch.

6·00 A. M.—Special details and idlers go ahead with work of the day as per port routine and orders.

6·00 to 8·00 A. M.—Pump out engine room, fireroom and shaft-alley bilges and clean bilge strainers. Boilermakers inspect boilers, and coppersmith water piping for leaks. Send up to blacksmith firing tools that need overhauling.

8·00 to 9·00 A. M.—Warrant machinist on watch inspect engine and firerooms, and relief warrant machinist remainder of engineer's department and see same ready for inspection. Have clothing and shoes in unauthorized places stowed away and take dry clothes off clothes lines in engineer's department.

6·00 to 8·00 P. M.—Sound all double bottoms and pump bilge double-bottom compartments dry. Close all watertight doors and manholes which are not to be used during the night. Chief water tender of watch give boilers in use light bottom blow, and light surface blow if water is greasy. Test water in boilers in use and in feed tanks with hydrometer and by tasting. Blacksmith see coal bunkers are not overheated and obtain temperatures if possible.

7·50 P. M.—Warrant machinist of watch report department secure in accordance with orders and routine.

Each watch will clean one fire in each boiler and blow tubes in one-fourth of all boilers in use in regular rotation. At five bells hoist ashes. (4·00 to 6·00 P. M. watch does not hoist ashes and 6·00 to 8·00 watch does not clean fires.) Use test cocks frequently and blow through all gauge glasses at least once each watch. Before being relieved pump out bilges, lay out coal for next watch and sweep up floor plates.

ROUTINE ORDERS.

(*To be kept at log desk in starboard engine room.*)

No. 1. July 30th, '06.'

Orders in this book will be copied in a smooth order book to be kept in the engineer's office, and the yeoman will see that each order is initialed in the smooth order book by all engineer officers and such petty officers as the senior engineer may direct.

All officers and petty officers will be responsible for the proper execution of these orders.

Officers or petty officers in charge of a watch will see that this order book is not lost or defaced.

(All officers and petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 2. July 30th, '06.

Attention is called to the port and sea routines posted in the bulletin board, which will be followed except when modified by the senior engineer.

(All officers and petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 3. Aug. 1st, '06.

When in port auxiliary watches will carry boiler steam pressure between 150 pounds and 175 pounds.

(All water tenders and petty officers who have charge of auxiliary watches initial.)

——— ———, *Sen. Eng. Off.*

No. 4. Aug. 1st, '06.

Whenever the ship is listed the list will be corrected by using coal and make-up-feed entirely *from the side towards which the ship is listed.* Boilers are not to be pumped down or up in order to correct the list.

(All water tenders and machinist's mates initial.)

——— ———, *Sen. Eng. Off.*

No. 5. Aug. 2d, '06.

Drinking-water buckets will not be used for any other purpose than drinking water; when not in use they will be turned into the storeroom, and storeroom keepers will see that they are kept properly marked "Drinking Water," in large letters with white paint.

(All water tenders, machinist's mates and storeroom keepers initial.)

——— ———, *Sen. Eng. Off.*

No. 6. Aug. 6th, '06.

Whenever water is taken out of or put into a reserve-feed double bottom, the fact will be noted in the log, thus: "Make-up-feed, B–99, 35 inches to 30 inches." "Filling boilers, B–98, 42 inches to 0 inches." "Distilling to B–94, 25 inches to 42 inches." "Water from water barge to B–95, 0 to 50 inches." It will be seen that this gives the reason why water is pumped in or out, and the soundings in inches both before and after pumping.

(All water tenders and machinist's mates initial.)

——— ———, *Sen. Eng. Off.*

No. 7. Aug. 10th, '06.

Hereafter there will be noted on the rough log sheet the amount of coal used for each watch from each bunker. The buckets of coal will be noted in this manner: "B–5, 10 buckets;" "B–7, 14 buckets;" "B–8, 22 buckets."

(All machinist's mates and water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 8. Aug. 10th, '06.

Whenever a coal bunker, double-bottom compartment, boiler or other confined space is opened, no man will be allowed to enter it until the air has been tested with a candle or bunker light. If the light goes out or burns dimly no one will be allowed to enter until the space is properly ventilated. Men at work in a confined space such as the above must be in constant communication with some one outside.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 9. Aug. 10th, '06.

When fires die out in any boiler the following work will be done as soon as possible:

(*a*) Give boiler light surface and bottom blow while pressure is on. Sweep tubes with steam hose. Open air cock when pressure is gone. Haul furnaces and ash pits, and overhaul grate surfaces. Renew any broken fittings. Overhaul gauge glasses and try cocks if they are out of order. Inspect

all manholes and handholes and seams, and stop leaks. Grind in leaky valves and remake leaky joints. See that dampers are working properly.

(*b*) If steam space is opened up examine zincs, and renew if one-half gone. Examine all interior piping and fittings which are accessible. Clean grease and scale off all surfaces.

(*c*) When through overhauling boiler, pump entirely full of fresh water, and close furnace and ash-pit doors. Test boiler with hydraulic pressure, if so instructed.

(All chief water tenders and boilermakers initial.

——— ———, *Sen. Eng. Off.*

No. 10. Aug. 10th, '06.

Whenever boilers, cylinders, feed tanks, condensers, etc., are to be entered for inspection or overhauling, all valves which would allow steam or water to enter must be secured by locking, lashing, or otherwise. Likewise, whenever anyone is working in a main engine cylinder, the turning engines will be secured.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 11. Aug. 12th, '06.

When raising steam in any boiler the following routine will be strictly followed:

(*a*) Pump boiler to steaming level.

(*b*) See no clothes or inflammable material around boiler or uptakes, and smoke-pipe covers off.

(*c*) See air cocks, steam-gauge valves and water-gauge valves are open.

(*d*) See all boiler-stop and surface and bottom-blow valves are closed.

(*e*) Start fires and report to engine room and officer of the deck that fires are started.

(*f*) See feed pump ready for use, and call a boilermaker who will look out for leaks and set up on joints in boiler if necessary.

(*g*) Report to engine room when steam forms.

(*h*) When steam has blown through air escape several minutes, close air cocks, test try cocks and blow through gauge glasses.

(*i*) Let fires burn slowly and evenly and do not force boilers without specific orders to do so.

(*k*) When pressure is about 100 pounds, try safety valves by lifting with hand gear.

(*l*) When steam pressure is a few pounds above that in the steam line, connect up boiler by opening up stop valve slowly.

(*m*) Report to engine room and officer of the deck when boilers are connected.

(All chief petty officers, water tenders and boilermakers initial.)

——— ———, *Sen. Eng. Off.*

No. 12. Aug. 10th, 06.

When turning steam into any steam line great care must be taken to avoid "water hammer." Drain valves on the steam line will be opened before admitting any steam, and the line will be heated up slowly by gradually opening the steam valve.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 13. Aug. 25th, '06.

At sea the yeoman will detail two men from each section, and water tender or other petty officer in charge, to throw down coal from the upper to lower bunkers. This coal-bunker detail will work during the day as per routine, and begin throwing down coal from the upper bunkers as soon as coal stops running out on the fireroom floor plates through lower bunker doors.

Bunker plates must not be left open when washing the upper decks or at night. When under forced draft care must be exercised that air does not blow through bunkers from the firerooms thus spoiling the draft and dirtying the upper decks.

(All water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 14. Aug. 25, 1906.

Water tenders in charge of a watch will, as soon as they come on watch, inspect all the firerooms to see that the routine and routine orders are being carried out, that fires are in proper shape, water at proper height in gauge glasses, feed pumps working properly, and everything about the boilers in good condition.

They will at sea allow no fires to be cleaned or tubes to be blown until there is over 175 pounds steam pressure, and then they will not allow too many fires to be cleaned at one time.

If necessary to keep steam pressure above 175 pounds, only one fire will be cleaned at one time.

Tubes will be properly blown and lances will be entered their full length between the nests of tubes.

(All water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 15. Aug. 25, 1906.

Clothing may be hung in the engineer's department only on clothes lines put up in the engine and firerooms by chief petty officers in charge of those compartments. Clothing left adrift will be turned into the master-at-arms.

(All petty officers initial.)

——— ———, *Sen Eng. Off.*

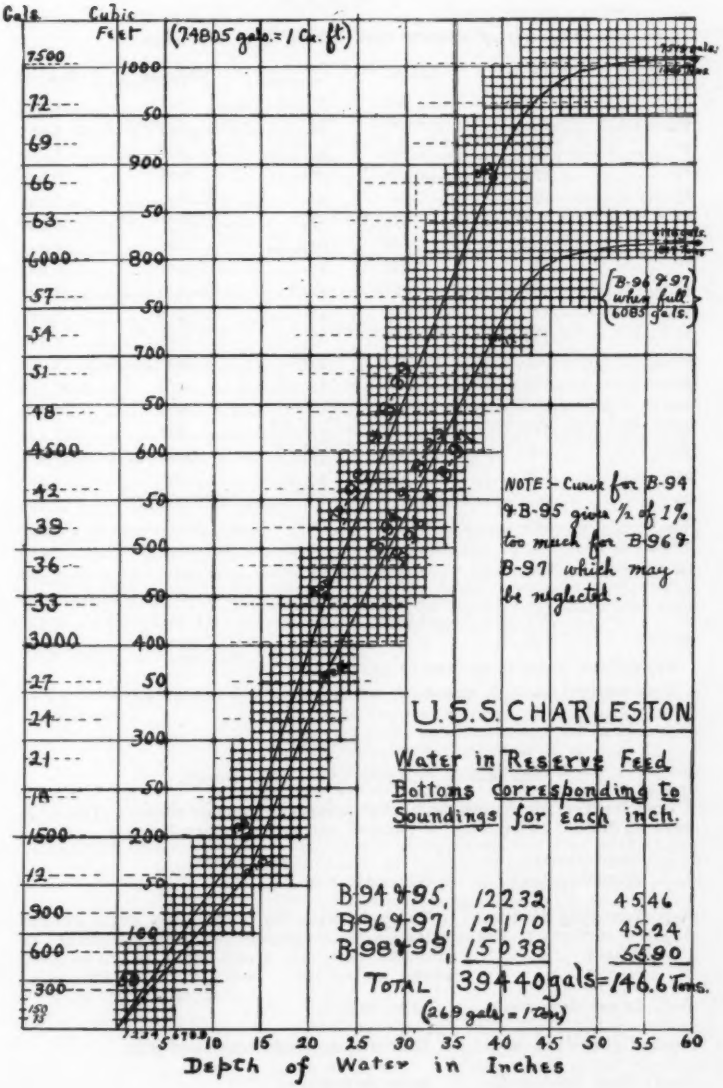

Gals.
Cubic Feet
(7.4805 gals. = 1 Cu. ft.)
7500
1000
72
50
69
900
66
50
63
6000
800
57
50
54
700
51
50
48
4500
600
42
50
39
500
36
50
33
3000
400
27
50
24
300
21
50
18
1500
200
12
50
900
100
600
300
B-96 & 97 when full 6085 gals.
NOTE - Curve for B-94 & B-95 gives ½ of 1% too much for B-96 & B-97 which may be neglected.
U.S.S. CHARLESTON
Water in Reserve Feed Bottoms corresponding to Soundings for each inch.
B-94 & 95, 12232, 45.46
B-96 & 97, 12170, 45.24
B-98 & 99, 15038, 55.90
TOTAL 39440 gals = 146.6 Tons.
(269 gals. = 1 Ton)
5 10 15 20 25 30 35 40 45 50 55 60
Depth of Water in Inches

Fig. 3.

U. S. S. *CHARLESTON.*

Capacity of Reserve-Feed Tanks, B-98 and B-99.

Soundings, in inches.	Cubic feet.	Gallons.	Soundings, in inches.	Cubic feet.	Gallons.
5	70	525	33	728	5,460
10	145	1,087	34	754	5,655
11	160	1,200	35	780	5,850
12	175	1,312	36	806	6,045
13	193	1,440	37	832	6,240
14	210	1,575	38	857	6,427
15	238	1,785	39	882	6,615
16	271	2,032	40	910	6,825
17	300	2,250	41	930	6,975
18	325	2,437	42	943	7,077
19	355	2,662	43	956	7,170
20	388	2,910	44	967	7,252
21	425	3,187	45	978	7,335
22	450	3,375	46	982	7,365
23	475	3,562	47	987	7,402
24	500	3,750	48	994	7,455
25	525	3,937	49	997	7,477
26	550	4,125	50	999	7,492
27	575	4,312	51	1,001	7,507
28	600	4,500	52	1,002	7,514
29	625	4,687	53	1,003	7,522
30	651	4,875	54	1,003½	7,526
31	677	5,077	55	1,004	7,530
32	703	5,272	56	1,005	7,537

269 gallons, equal to one ton (2,240 pounds).

Tons marked on D. B. manholes are not correct.

U S. S. *CHARLESTON.*

Methods of Pumping Out Compartments.

Coal bunkers, magazines and other compartments not shown in this table have no drains. They may be pumped out with a suction hose.

Compartments.	How drained.
A-1. Trimming tank.....	Sluices to A-2.
A-2. Trimming tank..... A-3. (3 suctions.) Forward hold.	Through man. No. 4, and sec. drain cut-out valve (in No. 1 fireroom), to *secondary drain;* or through man. No. 4 and man. No. 10 to *hand pump.*
A-5. Lower dyn. room...	Same as A-2.
A-20. Upper dyn. room...	Sluices to A-5—starboard and port.
A-97. Ford. hold D. B....	Same as A-2.
A-98. D. B. of 3-in. mag..	Same as A-2.

Compartments.	How drained.
A–99. Dyn. room D. B....	Double-bottom suction line to *auxiliary feed pumps, Nos. 2 and 3.*
A–117 and 118 chain lockers. (2 suctions.)	Same as A–2.
B–1, 2, 3 and 4. Fireroom 1, 2, 3 and 4.....	*Main or secondary drains* direct. Also direct bilge suction to *Auxiliary feed pump in each fireroom.* (Firerooms No. 1 and No. 2 sluice to D. B.'s under them.)
B–89, 90, 91, 92 and 93. Fireroom double bottoms.	*Auxiliary feed pumps No. 2 or No. 3* on double-bottom suction line. (There is a cut-out valve in suction line to auxiliary pump No. 2.)
B–94, 95, 96, 97, 98 and 99.	Reserve-feed tanks and connect to feed pumps.
B–109. Evaporator room..	Sluices to fireroom No. 4, starboard and port.
C–1. Starboard engine room.	*Stbd. M. circ. pump* direct bilge injection. *Main drain direct. Secondary drain direct* or on stbd. eng.-R. bilge line. *Stbd. F. and B. pump* or *shaft pump* on engine-room bilge line direct to bilge or crank pits. (There are sluices to D. B.'s under engine room.)
C–2. Port engine room...	Same as for starboard side.
C–97, 98 and 99. Engine-room double bottoms.	*Sec. drain* through man. No. 1 in port engine room and double-bottom line. *Port F. and B. pump* through man. No. 1 and double-bottom main. *Port shaft bilge pump* or *port F. and B. pump* through port engine-room bilge line, man. No. 1 and double-bottom suction line.
D–1. M. 3-in. magazine..	No drain.
D–2. Port shaft alley. (2 suctions.)	*Secondary drain* through man. No. 1 in port eng. room (cut-out valve in sec. drain near manifold) and port shaft-alley manifold in after part engine room. *Port shaft pump* or *port F. and B. pump* through port engine-room bilge line and shaft-alley man. (Cut-out valve on branch to engine-room bilge line.)
D–3. Stbd. shaft alley.....	*Sec. drain* through man. No. 2 in stbd. engine room and stbd. shaft-alley manifold aft in engine room. *Stbd. F. and B. pump* or *shaft pump* through shaft-alley manifold and stbd. engine-room bilge line. (Cut-out valve in branch to engine-room bilge line.)
D–4 and 5. Shaft alley wing compartments.....	No drains.
D–6. Wing D. Bottom...	*Sec. drain* through man. No. 3 (aft in port engine room), after bilge line and man. No. 5 in port shaft-alley. *After hand pump* through hand-pump man. (in P. eng. R.) man. No. 3 and man. No. 5, as above.

Compartments.	How drained.
D–7. Wing D. Bottom...	*Sec. drain* through man. No. 3 (aft in P. eng. R.), after bilge line and man. No. 6 in after hold. *After hand pump* through hand-pump manifold and manifold No. 3.
D–8. After hold.....	Same as D–7.
D–9. Trimming tank.....	*Sec. drain* through man. No. 3 (P. eng. R.) or *after hand pump* through pump manifold and man. No. 3.
D–10. Trimming tanks...	Sluices to D–9.
D–98 and 99. 3-in. magazine double bottoms.	Same as D–6.
D–25 and 26. Steering engine flats.	Sluice to D–8. (After hold.)

Pump Connections, Main Drain.

Pumps.	How connected to main drain.
Starboard main circulating.	Open main-drain suction and overboard discharge at pump.
Port main circulating.....	Same as starboard.
No. 1, auxiliary feed.....	Suction valve at each pump and cut-out valve in branch to main drain in each fireroom.
No. 2, No. 3 and No. 4, auxiliary feed.	Same as No. 1 auxiliary feed.
Starboard engine-room fire and bilge.	Man. No. 2 abaft pump and cut-out valve in branch to main drain (in bilge ford. in starboard engine room).
Port eng.-room fire and bilge.	Man. No. 1 abaft pump and cut-out valve in starboard engine room as above.
Starboard shaft pump.....	Starboard engine-room bilge line, man. No. 2 and cut-out valve as above.
Port shaft pump.....	Port engine-room bilge line, man. No. 1 and cut-out valve as above.
After hand pump.....	Bilge suction of after hand pump man. (in P. engine room), man. No. 3 (aft in port engine room), man. No. 1 (in port engine room) and cut-out valve as above.

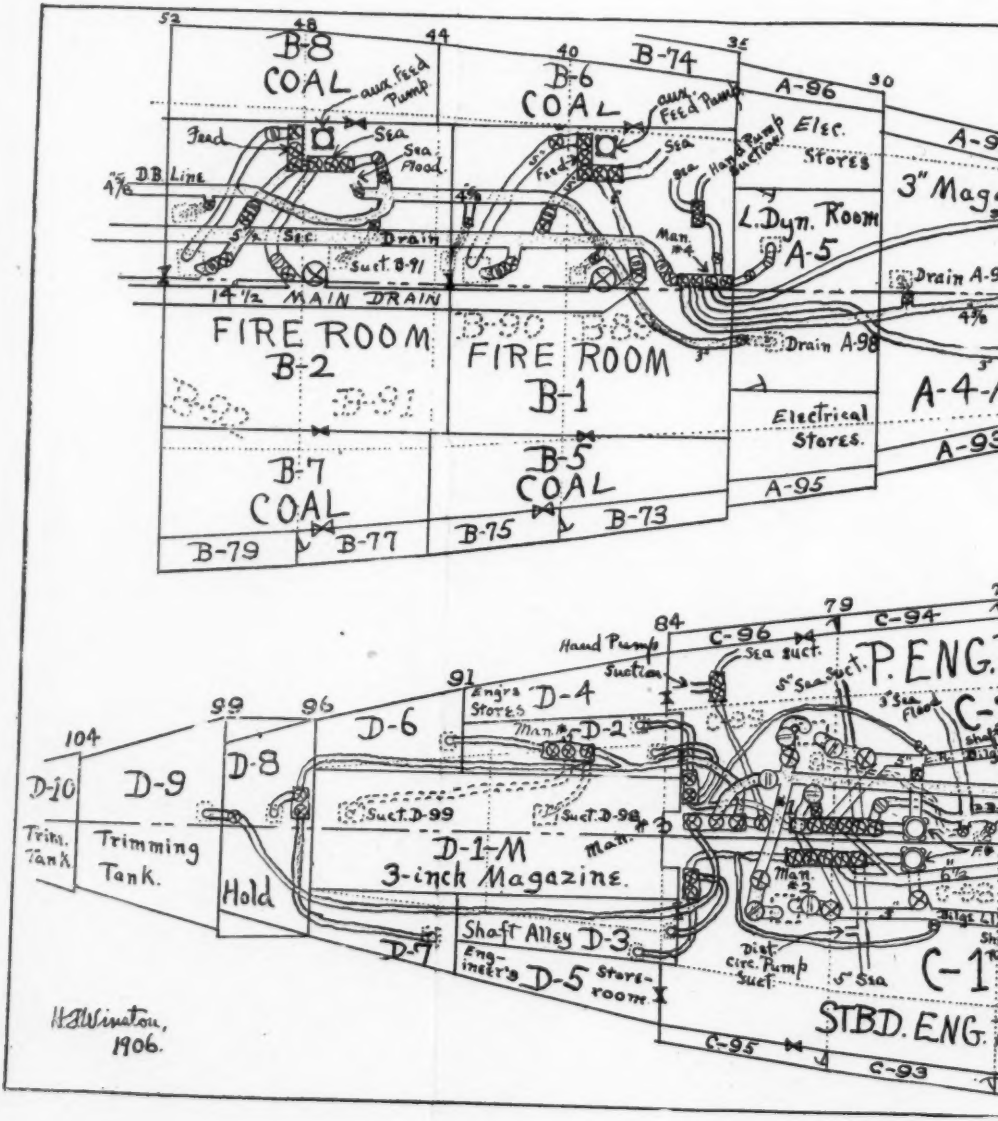

B-8
COAL
B-6
COAL
B-74
A-96
Elec. Stores
3" Mag
L. Dyn. Room
A-5
Aux. Feed Pump
Sea
Sea Flood
Feed
DB Line
Suct. B-91
Drain
14½ MAIN DRAIN
FIRE ROOM
B-2
FIRE ROOM
B-1
Drain A-98
Electrical Stores.
A-95
B-7
COAL
B-5
COAL
B-79
B-77
B-75
B-73
Hand Pump Suction
Man. #4
Drain A-9
A-4-
A-9
B-90
B-91
B-88
B-89
104
99
96
91
84
79
D-10
D-9
D-8
D-6
D-4
C-96
C-94
P. ENG.
Trim. Tank
Trimming Tank.
Hold
Angrs Stores
Man. #2
D-2
Sea Suct.
5" Sea Suct.
3" Sea Flood
Suct. D-99
Suct. D-98
D-1-M
3-inch Magazine.
Main
Shaft Alley D-3
D-7
Engineer's D-5 Storeroom
Dist. Circ. Pump Suct.
5" Sea
C-1
STBD. ENG.
C-95
C-93
H. H. Winston, 1906.

Fig. 4.

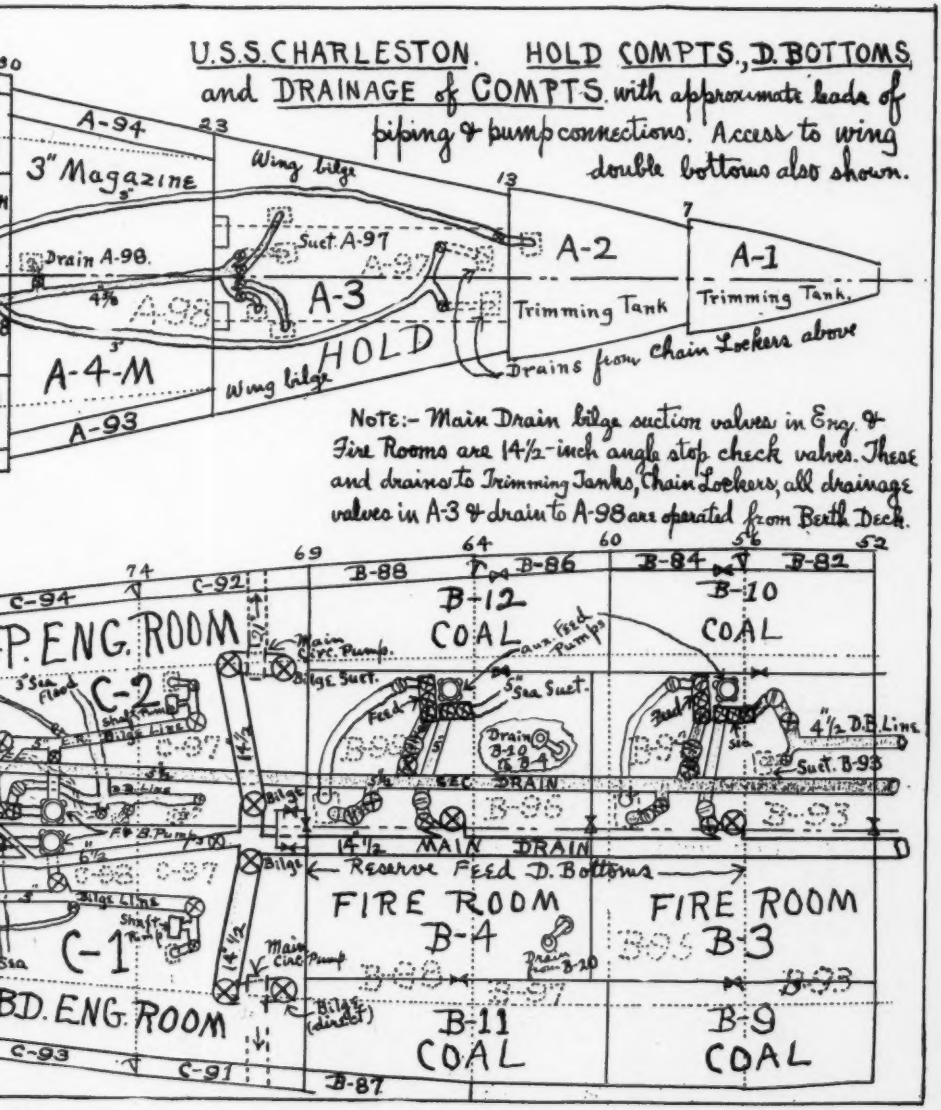

U.S.S. CHARLESTON. HOLD COMPTS., D. BOTTOMS, and DRAINAGE of COMPTS. with approximate leads of piping & pump connections. Access to wing double bottoms also shown.
A-94
3" Magazine
Wing bilge
Drain A-98.
Suct. A-97
A-3
A-2
A-1
Trimming Tank
Trimming Tank.
HOLD
A-4-M
Wing bilge
A-93
Drains from chain Lockers above
NOTE:- Main Drain bilge suction valves in Eng. & Fire Rooms are 14½-inch angle stop check valves. These and drains to Trimming Tanks, Chain Lockers, all drainage valves in A-3 & drain to A-98 are operated from Berth Deck.
C-94
C-92
B-88
B-86
B-84
B-82
B-12
COAL
B-10
COAL
P. ENG. ROOM
C-2
Main Circ. Pump.
Aux. Feed Pumps
Bilge Suct.
5" Sea Suct.
Feed
Drain B-10 & B-84
4½" D.B. Line
Suct. B-93
14½ MAIN DRAIN
Reserve Feed D. Bottoms
FIRE ROOM
B-4
FIRE ROOM
B-3
C-1
BD. ENG. ROOM
Main Circ. Pump
Bilge (direct)
B-11
COAL
B-9
COAL
C-93
C-91
B-87

Fig. 4.

Pumps.	How connected to main drain.
Dist. circ. pump.....	Close sea valve at ship's side (stbd. eng. room), open "sea" valve in man. No. 2 (stbd. engine room), main drain valve in man. No. 2 and cut-out valve as above.

Pump Connections, Secondary Drain.

Secondary drain connects to main drain through a branch in starboard engine room (branch has a cut-out valve) and manifolds No. 1 or No. 2. Consequently main drain pumps may be used in this way on the secondary drain.

Pumps.	How connected to secondary drain.
No. 1, No. 2, No. 3 and No. 4 aux. feed pumps..	Suction valve at each pump and cut-out valve on branch to secondary drain in each fireroom.
Forward hand pump.....	Hand pump man. in No. 1 fireroom, man. No. 10 in same fireroom and cut-out valve to secondary drain (just abaft man. No. 10).
After hand pump.....	Hand pump man. (in port engine room), hand pump suction in man. No. 3 (port engine room) and secondary drain valve in man. No. 3.
F. and B. pump, starb'd engine room.	Direct suction through man. No. 2 in starboard engine room, or through engine-room bilge line and man. No. 2.
F. and B. pump, port engine room.	Same as above, except man. No. 1 in port engine room.
Starboard shaft pump.....	Starboard engine-room bilge line and man. No. 2 in starboard engine room.
Port shaft pump.....	Port engine-room bilge line and man. No. 2 in port engine room.
Distiller circ. pump.....	Close sea valve at ship's side in starboard engine room. Open "sea" valve in man. No. 2 and secondary drain valve in same manifold.

APPENDIX B.

DUTIES OF YEOMAN AND ASSISTANTS.

Daily, before quarters, correct lists of absentees, sick, working parties, etc., in engine room, and at same time take down obsolete lists and notices.

Bring to quarters lists of excused absentees.

Before noon make up coal report and fill in "bunker expenditure book;" write up and have signed log book

(reading water meters to see that gallons of water distilled "for feed" and "for ship" are correct, and read scale on gauge glasses of oil tanks for correct amount of oil "expended" and "on hand").

About 11·30 A. M. set all clocks of engineer's department, obtaining correct time from chief quartermaster.

At noon take log and coal account to senior engineer's room for his signature.

At 7·30 P. M. take "daily order book" to chief for orders of the next day.

At end of week enter up expenditures of stores as per rough books of storeroom keepers. Bring up to date smooth copy of log to be sent to Washington.

At end of month correct and bring up to date watch, quarter and station bills and all bulletin-board notices. General straightening up of everything in engineer's office. Submit to captain monthly report of expenditures and receipts of stores.

Quarterly send in smooth copy of steam log and quarterly reports (page 381, Navy Reg.) to Navy Department. Make out surveys for stores, Title "B," which are lost or damaged.

Semi-annually (June and December), make out requisitions for six months' supplies, Title "Y," and such articles as have been condemned by survey under Title "B." Correct "spare-part book" and requisition for spare parts used up. (This book contains a list of the spare parts of machinery on board and where each is stowed.)

Annually (December,) take annual inventory of stores.

Requisitions for stores on allowance book, of which the amounts are "as required," such as oil, waste, grate bars, etc., must be made out when they run low.

Books of supplies and accounts will be kept as per Art. 1546, Navy Regulations.

Copies of all official letters written will be kept in a book marked "Letters Sent," and originals or copies of all letters received will be kept in a book marked "Letters Received."

The "Coal-bunker expenditure book" will be made out in accordance with the following form:

Lower bunkers.....	B–5	B–6	B–7	B–8	B–9	B–10	B–11	B–12	Totals.
Upper bunkers.....	B–101	B–102	B–105	B–106	B–111	B–110	B–115	B–114	
	79	79	112	112	123	123	126	126	
	103	103	100	100	102	102	100	100	
Capacities	182	182	212	212	225	225	226	226	1690
Sept. 30, on hand..	182	182	212	212	225	225	226	226	1690
Oct. 1. Expended..	3 220	...	...	...	19 80	24 320	30	...	76 870
Remaining	178 2020	182	212	212	205 2210	200 1920	196	226	1614 1670

A rough note book marked "Repairs Needed," will be kept hanging in a conspicuous place, in which officers of the department may note desirable repair work, and from which "necessary repairs" may be submitted to the captain on arrival in port.

It will be noted that at the top are the lower bunkers with the bunkers which trim into them just below. Then come the capacities of the bunkers and of each pair of bunkers. In making up the noon coal report it is usual to add together the expenditures for each watch during the preceding 24 hours. The coal-bunker book is filled in by entering the expenditures from each bunker (or pair, upper and lower) for the preceding 24 hours. Then the "total remaining," in last column, should agree with the noon coal report and is a check on it.

By keeping this separate coal account for the various bunkers there is always on hand a check for the amounts estimated in each bunker by inspection. In addition, whenever a bunker is emptied the bunker book shows at once if the proper amount of coal is being expended each day. If not, either the number of buckets turned in is incorrect or the number of pounds allowed per bucket is wrong, and the error may be corrected before there is a large discrepancy.

Delicate instruments, such as drawing instruments, micrometer gauges, indicators, gauge-test outfits, etc., will be kept in the log room, and the yeoman will be responsible that they are so stowed that they will not be broken.

Drawings will be taken from the log room only by an officer, and he will place his initials opposite the number of the drawing in the "Index Book" of drawings when the drawing is taken out. Engineer's pocket books and other books designated will not be taken from the office.

Text books may be taken by any member of the engineer's force for one week, and will be charged by the yeoman when taken out and the date noted.

Keys from the key board will be delivered only to an officer or person who is responsible for the compartment or mechanical device to which the key belongs.

The log room will be locked and the key kept by yeoman, or turned in to the senior engineer officer when none of the yeoman force are there.

When getting under way or coming to anchor there will be a yeoman or one of his assistants at each engine-room desk to make notes for the log.

DUTIES OF STOREROOM KEEPERS.

The senior or head storeroom keeper has general charge of all the storerooms and superintends the stowage of stores and their issue, and is responsible for the care, preservation and proper expenditure of stores.

All storeroom keepers will stand watch with their sections, or as the senior engineer may direct, and will observe the rules prescribed for storeroom keepers.

The senior will keep all storeroom keys and see all storerooms locked whenever no storeroom keeper is on watch. In case of his absence from the ship these keys will be turned over to the next storeroom keeper in seniority or to the engineer's yeoman.

Stowage.—As far as possible all stores of the same kind will be stowed together and in such a manner that a portion of each kind of stores are easily seen. Stores of one kind must never be entirely hidden by stowing them behind other stores. Articles wrapped up in packages, or in boxes, must be properly marked or tagged, with the marking in plain view.

Articles of steel or iron must not be stowed in damp places where they will rust.

Rubber goods and composition packings must not be stowed in a hot, dry place, where the rubber soon drieds up and becomes brittle.

All instruments and other articles which may be easily broken (steam gauges, thermometers, etc.) must be stowed carefully where they will not be injured when the ship is at sea.

Spare parts of machinery, such as heavy bolts and nuts of steel, will be painted with white lead to prevent rusting. Other spare parts of steel or iron will be covered with white lead and tallow or vaseline (when bearing surfaces). All spare parts likely to be injured will be wrapped with canvas in order to protect them.

Once a quarter (in March, June, September and December) all spare parts will be examined and overhauled if necessary. Whenever spare parts are used or their stowage changed the yeoman is to be notified so he may correct his "spare-part book."

In December the annual inventory of stores will be taken.

Books.—A charge book for tools, instruments, etc., will be kept in each storeroom, and such articles will be charged against the person taking them out. When the day's work is completed these tools, etc., must be returned to the storerooms, and before 8·00 P. M. each day the storeroom keeper on watch will report to the officer on duty whether all are returned or not.

No tools or stores will be issued to any one outside of the engineer's department, except by order of the senior engineer.

In a rough expenditure book will be entered the stores expended each day, and at the end of the week this book will be turned in to the yeoman, who will copy the expenditures into his book.

There will also be kept a list of stores, Title "B," which, having been used, damaged or lost, must be surveyed.

Stores such as oil, waste, paint, packing, canvas, etc. (which are in Title "Y"), will be issued only upon the order of an officer of the department.

Drinking-water buckets will be served out when making preparations for going to sea, and returned to storerooms upon arrival in port.

The storeroom keeper will see that they are properly painted a light blue and lettered in large white letters "Drinking Water," before serving them out. All wash buckets will be painted with a broad green stripe before being served out.

Men who come to storerooms for stores must wait outside, and have same served out to them by the storeroom keeper.

Wash-Room Rules.—Each wash-room keeper will give the paint work, decks, bowls, etc., in his wash room a thorough scrubbing each day before morning quarters. All men must be clear of the wash room twenty minutes before quarters so it may be properly cleaned for inspection.

Wash rooms will also cleaned up after dinner and after 8·00 P. M., as soon as the watch finishes washing up.

Wet clothing will not be allowed to be hung in wash rooms.

When leaks are discovered in the water piping, or drain pipes are plugged, the plumber will be notified, and if he does not repair the defects, the fact will be reported to the officer on duty. Anyone who fails to obey the above rules will be reported to the officer on duty in the engine room.

Rules for Steam Launches.—Whenever the order is received from the officer of the deck to secure the launch, the boiler will be given a light blow by opening the blow-down valve wide and then closing it again immediately. At the same time the boiler tubes will be swept with a steam lance, if there is no fresh paint work near the launch, which will be soiled.

When raising steam the safety valve will be lifted by hand to see that it does not stick, and water gauges blown through and try cocks tested.

If it is necessary to feed salt water into the boiler at any time the fact will be reported to the officer in charge of auxiliaries as soon as possible, and the boiler will be opened up and cleaned shortly thereafter.

Once a quarter (in January, April, July and October) all

the launch machinery will be opened up, examined and overhauled. This includes the interior of boiler, cylinders and valve chests of engine, air pump, feed pump, water tanks, coal bunkers, bilges, and supports for engines and boilers, and the keel condenser. The settings of valves and clearances of pistons will be taken, but will not be altered except under the direction of officer in charge of auxiliaries.

Whenever the launch is to remain in its cradle for five days or more the cylinders will be opened up, wiped out and lightly coated with vaseline.

Spare parts of machinery in the launch will be kept covered with white lead and tallow or vaseline, and examined frequently to see that they are not rusting. When spare parts are used the fact will be reported to the yoeman so his books may be kept straight.

Machine-shop Rules.—The machinist in charge will be responsible for the good condition of the machine shop and the care and preservation of all machines and tools there.

When he and his helper (fireman or coal passer) are absent from the machine shop he will lock the door and keep the key, or, if he is to leave the ship, turn it in to the engineer's office. He will keep a charge book or slate and charge up tools to the person taking them out. If these tools are not returned by 8·00 P. M. he will report the fact to the officer on duty.

No one will be allowed to stow clothes or other articles in the machine shop at any time except the machinist in charge and his helper, who will be allowed to stow personal belongings only in such space as may be assigned them.

A list will be taken of all stores or tools which have been expended, lost, damaged or worn out, and handed into the yeoman each Saturday morning to be entered on his books.

Rules for Evaporator Room.—Whenever any evaporator is started or shut down the fact will be reported to the engine room.

In starting up an evaporator, water will be discharged to the

reserve-feed tanks until it is known by testing and tasting the water that it is perfectly fresh.

When running an evaporator, the water level will always be low enough to leave the upper row of the nest of steam tubes bare. Shell pressure will be kept at from 6 to 8 pounds, and should be even lower when starting up an evaporator that has just been scaled.

Every four hours, at seven bells, the evaporator in use will be blown down, and when empty will be quickly filled to steaming level with cold water before admitting any steam. Then the steam valve will be opened and steam rapidly admitted to the nest of tubes. This quick cooling and then heating of the tubes causes them to first contract and then expand, and cracks off the scale.

Great care must be taken that idle evaporators are kept dry to prevent rapid corrosion and rusting. For this reason particularly, all valves which would allow water or vapor to back up into the shell must be kept closed tightly, and these valves renewed or ground in as soon as they begin to leak.

After six weeks' continuous use evaporators will be scaled and interior of shell thoroughly examined. Scale need not be removed from the shell, as it protects the surface of the metal. Interior piping must be examined to see that it is not plugged up or corroded, and the slits in dry pipe freed of any deposit.

APPENDIX C.

MONTHLY ROUTINE.

Obtain some indicator cards of main engines.

Senior engineer inspect all compartments, watertight doors and mechanical devices and submit written report to captain.

Inspect auxiliary steam machinery under other bureaus.

Clean out feed and filter tanks and wash or renew filtering material.

Test fireroom blowers.

Scale evaporators that have been in use and renew zincs, if one-half gone.

Inspect steam-launch machinery. (Interior of boiler, safety valves, check valves, pumps, engine, shafting and keel condenser.)

Overhaul oil service of main engines.

QUARTERLY ROUTINE.

Examine holding-down bolts and all nuts about the framing of main engines.

Examine valves of main air pumps, of auxiliary air and circulating pumps, and the steam cylinders and valves of these pumps.

Boil out main and auxiliary condensers with soda.

Open up main-engine cylinders, examine piston rings and springs, and measure clearances.

Open up main-engine valve chests, examine piston rings and clean balance pistons.

Try relief valves on main cylinders and jackets and overhaul jacket reducing valves.

Paint engine and firerooms.

Inspect and overhaul all spare parts of machinery, tools and wrenches.

Test steam gauges on engine-room gauge boards, of boilers and important gauges of auxiliary machinery and steam lines.

Examine main shafting, bearings, stuffing boxes and propellers as far as possible.

Examine zincs in zinc boxes of salt-water piping, and renew if one-half eaten up.

Overhaul boiler-feed and hotwell pumps.

Test safety, relief and reducing valves of auxiliary steam piping and auxiliaries of department.

Open up boilers, clean interior parts, examine interior fittings, and renew zincs if one-half gone.

Set safety and sentinel valves of boilers if not correctly set.

Clean air space between uptake and boiler casing.

Overhaul uptake dampers.

SEMI-ANNUAL ROUTINE.

Open up all condensers and examine tubes.

Examine shafting and interior of water chamber of main circulating pumps.

Examine valves in all manifolds.

Go over main and auxiliary exhaust piping and remake joints where leaking.

Grind in or renew valves in steam lines which ordinarily cannot be taken out on account of constant use.

ANNUAL ROUTINE.

Test compressed-air system to full pressure.

Take yearly inventory of stores.

SPECIAL OCCASIONS.

Upon Arrival in Port.—Report of repairs necessary (Art. 669, *Navy Reg.*).

Obtain coal, oil, water and waste, if needed.

Very Cold Weather.—Keep piping in steam launches from freezing. Either keep steam up and turn over engines frequently or drain the machinery entirely, including the keel condenser.

In Dry Dock.—Arts. 1638–9, Navy Reg.

Dock Trials.—Art. 1601, Navy Reg.

Speed Trials.—Arts. 1602 to 1605, Navy Reg.

BOOKS AND NOTES FOR REFERENCE.

Barton's "Naval Engines and Machinery."

Bieg's "Naval Boilers."

Liversidge's "Engine Room Practice."

"Engineer's Pocket Book." (Kent's is a good one.)

"Carpenter's "Experimental Engineering."

Durand's "Practical Marine Engineering."

Neilson's "Steam Turbine."

Welsh's "Ship and General Painter's Hand-book."

(The above usually found in library on board ship.)

Bureau of C. and R. "Specifications" for the ship.

Bureau of C. and R. "Booklets of Plans and General Information."

Typewritten description of drainage and fire systems.

(Usually found in executive officer's office.)

Bureau of S. E. "Machinery Specifications."

Bureau of S. E. "Booklets of Pipe Plans."

"Synopsis of Official Four-hours' Trial" of ship.

"Speed, Power and Revolution Curves." (Standardization trial.)

(Usually found in engineer's office.)

The following are not found on board ship, but may be obtained from files of the "JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS," or, in the case of patented devices, generally by simply writing the manufacturers:

Babcock & Wilcox Company's "Marine Steam."

W. W. White's "Coal Consumption of Main and Auxiliary Machinery of the U. S. S. *Minneapolis*." (JOURNAL of A. S. N. E., Vol. X, No. 1.)

B. C. Bryan's "Method of Calculating the Coal Endurance and Steaming Radius of War Vessels Under Various Conditions." (JOURNAL A. S. N. E., Vol. XVII, No. 3.)

H. T. Winston's "Selection and Use of Certain Apparatus and Material." (JOURNAL A. S. N. E., Vol. XVII, No. 2.)

Description of ship and her trials as written up in the JOURNAL of A. S. N. E.

Description and notes on the care and management of special, patented machinery on board; such as air compressors, ice machines, reducing valves, governor valves, steam traps, anchor engines, steering engines, speed indicators, boiler time-firing devices, engine indicators, machine-shop tools and machines, etc. For example, the Leslie Company have issued "Instruction Book, No. 2" (for their reducing valves), and the Crosby Steam Gauge and Valve Company, "Practical Instructions for using the Steam Engine

Indicator." A catalogue of a dealer in engineering sorets will also be found useful, such as Montgomery Ward & Company's (New York) "Tool Catalogue."

U. S. S. *CHARLESTON.*

Theoretical Coal and Water Consumption with Natural Draft.

Sept. 19th, 1906. H. T. WINSTON.

Revs. of engines.....	70	80	90	100	110	120	130	From trial-trip curves.
I.H.P. of engines....	3,000	4,300	6,100	8,400	11,200	14,500	18,500	
Speed in knots.....	12	13.6	15.1	16.6	18.1	19.45	20.6	
Tons coal per day...	94	140	193	270	365	470	...	3 lbs. per I.H.P.
Tons coal per day...	73	103	140	188	248	321	410	2 lbs. per I.H.P.
Tons coal per day... (See note.)	68	96	133	183	234	303	388	1.8 lbs. per I.H.P.
Tons water per day. ¼ of coal (2 lbs. per I.H.P.)	18	26	35	47	62	80	102¼	

NOTE.—First entry shows coal per day at rate of 3 pounds per I.H.P. of main engines per hour plus 10 tons per day for auxiliaries. This is approximately the amount most vessels of the U. S. Navy burn.

The second coal entry shows the coal per day, assuming 2 pounds per I.H.P. per hour plus 10 tons for auxiliaries per day. This is a good showing, but should be expected with good coal and normal conditions.

The third coal entry shows coal per day at the rate of 1.8 pounds per I.H.P. plus 10 tons per day. It is probable that the coal consumption will always be greater than this, and a smaller coal consumption will indicate that the coal account is incorrect.

Water expenditure in tons per day is the extra water required by the boilers when using 2 pounds of coal per I.H.P. per hour (normal consumption).

CORROSION OF PROPELLER SHAFT, U. S. S. *RHODE ISLAND.*

By Henry E. Rhoades, Passed Assistant Engineer, U. S. N., Member.

These notes and the accompanying drawing, Plate I, are submitted, not because the writer assumes that he has made a great discovery, nor to exploit what he does (not) know about electrical science, but that (possibly) what may be presented here will give reflection and study upon a condition disclosed on one of the outboard parts of the steam machinery of one of the new warships, and the further investigation of which may prove of value and benefit to those who take interest in it.

The incident referred to is that of the starboard main propeller shaft of the battleship *Rhode Island*, recently built by the Fore River Shipbuilding Company, and its condition was discovered and reported upon by a special Board of officers appointed to inspect the ship while in dry dock at the Navy Yard, Boston, Mass., just after arrival from the works of the builders to be transferred to the Government, a little more than a year ago, in October, 1905. The Board was not able at that time to make the most thorough examination of the shaft, such as was recently made, hence the first impression was that the groove, shown by the spiral line, might have been caused by ordinary corrosion, or by a scratch by some obstruction during the dock trial. The more minute examination subsequently made showed that the groove was distinctly cut for a length of about 56 inches, extending in the form of a right-hand spiral, from a point a foot from the fairwater on the propeller strut forward to about 4 feet 4 inches from the fairwater, and running about two-thirds the circumference

around the shaft. This groove was about 1½ inches in width at the widest point, and its depth ranged from ⅛ to $\frac{15}{32}$ inch throughout its length, the deepest point being where the groove was the widest.

In addition to the groove, and indicated by the smaller spots on the print both forward and abaft of the spiral groove, there were counted 292 distinct pittings extending in a direct line with the axis of the shaft and within a surface about two-thirds the circumference of the shaft, so that fully one-fourth, or more, of the circumference of the shaft remained scatheless. Fifty-one of these pits were ⅛ to ¼ inch deep, and thirteen from ¼ to ⅜ inch deep, while the others were less than ⅛ inch deep. The pits were generally irregular in form, although many of them were cup-shaped or dished. As the shaft was 17 inches in diameter it will be observed that these pittings were confined to about two-thirds the circumference, none being outside of the fore-and-aft boundaries and all within the limit of the two ends of the spiral groove. The estimate made by the Board as to the loss of strength from the original shaft through the decomposition that had taken place prior to the ship being turned over to the Government, resulted in a recommendation that the contractors substitute a new shaft, to which proposition they demurred, upon the claim that the designed strength of the shaft as originally called for had not been diminished. The Bureau of Steam Engineering sustained the recommendation of the Board, however, and a new shaft was supplied and installed in the ship in December, 1906. It must be admitted that this action of the officers of the Navy was justified by all the conditions met, and that the Government has obtained no more than what was its just due.

A casual glance of the groove after the shaft had been removed suggested that it might have originated from carelessness while being forged; that it might have been due to the one in charge of the drop-hammer allowing his scaler, an iron rod, to which is attached the spatula for keeping the scale removed from the surface of the metal, to be caught

under the hammer and welded with the steel. The rod being of iron and the shaft of steel, the action of salt water against the different densities of metal would excite corrosion and in the due course of time the rod would become dislodged. But a more careful inspection of the groove, with the aid of a microscope, disclosed distinctly lines such as would be made by a line of twisted wires of, perhaps, three-quarters of an inch in diameter.

A further investigation of the groove and the pittings of the shaft led me to the conclusion that they were produced by electrolysis, that process of galvanic action which is one of the conditions of electro-chemical decomposition. This galvanic action was caused, no doubt, by bringing into contact two dissimilar metals—the steel shaft and the strands of copper wire —the action being further excited by the action of the salt water under the wire around the shaft. When the vessel was put into dry dock, the hypothetical wire rope had disappeared, it probably having become dislodged from its long resting place by the revolution of the shaft when steaming from the Fore River Iron Works, at Quincy, to the Navy Yard, Boston; but the distinctively marked outlines of the strands of the wire rope in the corroded groove were sufficient evidence that a length of copper-wire rope, presumably electric wire, had accidently, or carelessly, been dropped overboard and lodged itself over the shaft where it had remained for months. The length of the spiral gutter or groove in the shaft indicated that both ends of the wire rope had hung downward from the shaft and, possibly, grounded on the bottom.

It was apparent that that part of the shaft most seriously pitted was that uppermost, or nearest the surface of the water, while the decomposition was in progress. By the contact of the two dissimilar metals—the steel shaft and the copper wire —they were charged with different electricities. The series of discharges that may be transmitted in a wire connecting a prime conductor (in this instance the steel shaft) of a machine in action with the ground possess the characteristics of a galvanic current, and these discharges were likely intensified

by the large mass of steel together with the copper-wire rope, and, probably, further excited by the composition sleeve on a part of the shaft.

It is a well-known fact that chlorine, which is the negative element of sea water, in itself decomposes metals with which it may be brought into contact; and the chemical energy of that liquid in contact with the dissimilar metals referred to, facilitated the discharge, and the current being thus maintained the electro-chemical decomposition was promoted. It is possible, also, that this decomposition, or electrolysis, may have been helped by the large percentage of manganese in the steel shaft; but no borings were taken, hence no determination was reached upon this proposition.

Whether or not these assumptions are correct, I concluded that the incident was so unusual and remarkable as to be well worth exploiting, because it may prove a text for further investigation of similar possibilities in the handling of warships.

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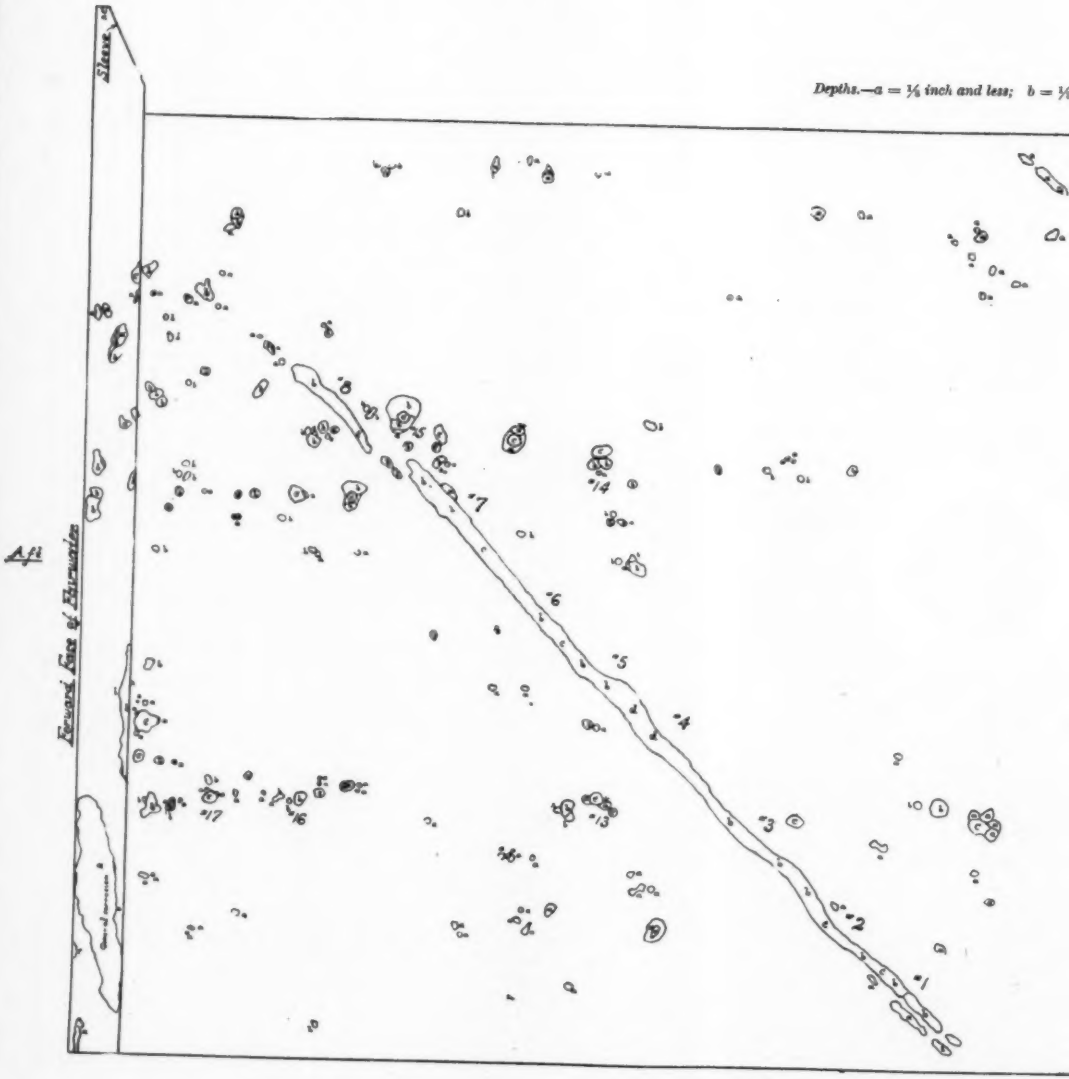
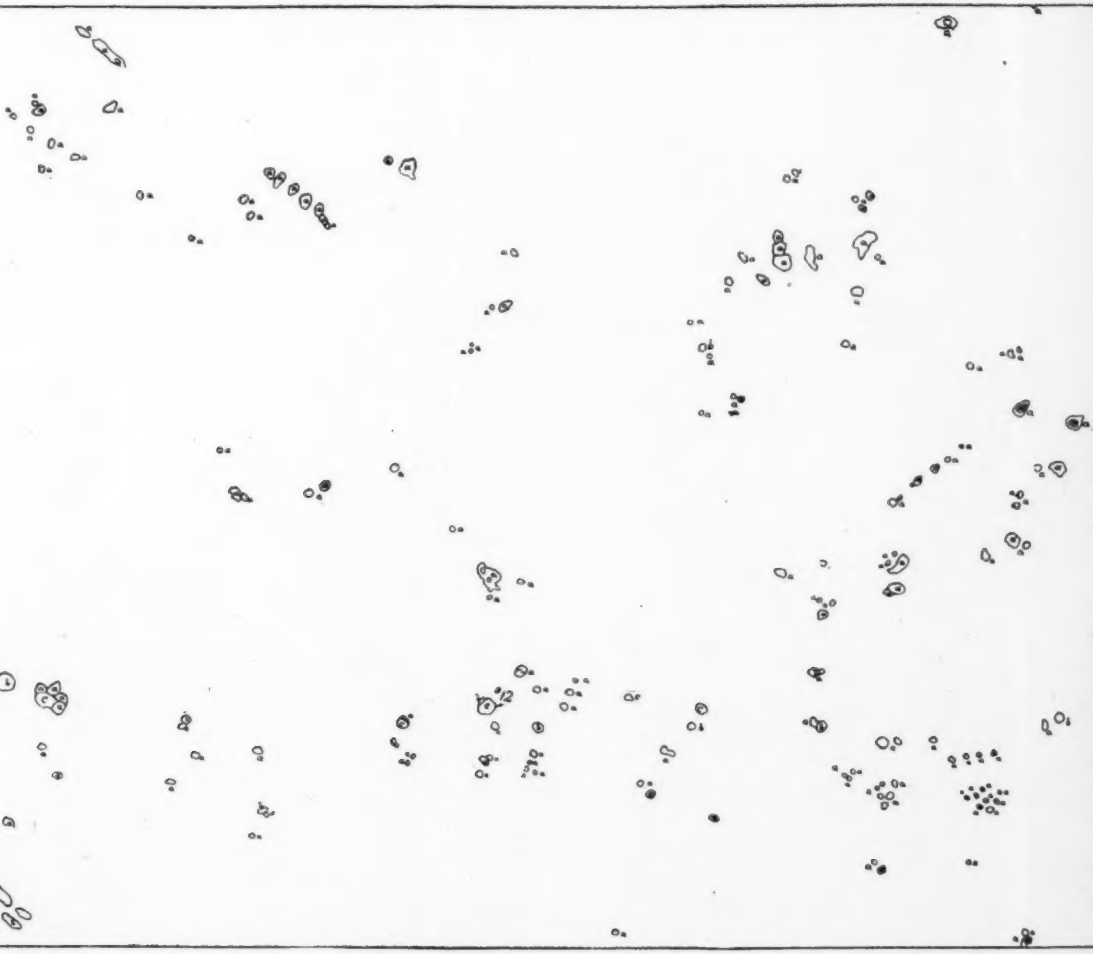


Plate I.—Development of Shaft B

h and less; b = ⅛ inch to ¼ inch; c = ¼ inch to ⅜ inch; d = ⅜ inch to ½ inch.



NT OF SHAFT BETWEEN BEARINGS, U. S. S. "RHODE ISLAND."

Depths.—a = 1/8 inch and less; b = 1/8 inch to 1/4 inch; c = 1/4 inch to 3/8 inch; d = 3/8 inch to 1/2 inc

Plate I.—DEVELOPMENT OF SHAFT BETWEEN BEARINGS, U. S. S. "RHODE ISL

$d = \frac{3}{8}$ inch to $\frac{1}{2}$ inch.

"RHODE ISLAND."

STANDARD LIGHTNING PROTECTION FOR THE CONSOLIDATED POWER-PLANT CHIMNEYS AT UNITED STATES NAVY YARDS.

By N. Monroe Hopkins, Ph. D.,

Electrical Engineer for Consolidated Power Plants, Department of the Navy.

In the design of a modern central light and power plant for military purposes reliability of operation is the first essential. There are many factors and influences, great and small, which tend to interrupt service, and they should all be carefully taken into account by the designing engineer. A breakdown of one of the new central power stations, or even a temporary failure to supply the important shops of construction and repair, equipment, ordnance and engineering, or the great electric pumps for emptying the dry docks, might, at a critical time, mean the defeat of a fleet. Insomuch as minute attention is given to the design of the machinery in duplicate and in triplicate, according to the best and most approved engineering practice, the work would be incomplete if the buildings and chimneys were not given the best protection from violent lightning strokes that modern practice can afford.

It has been with the view of providing chimney protection commensurate with the importance of the power plant to the naval service, that the writer took up the design of the lightning protection herein described. There are few devices in the field of applied electricity or electrical engineering regarded with greater skepticism and distrust than the lightning rod, but unfortunately so, for its reputation has been based in the majority of cases upon the old incorrectly applied con-

ductors of Benjamin Franklin's day. The early rods were put up with the idea that a lightning stroke took the form of a direct-current flash to earth, that it was attracted and influenced by the presence of iron or steel, and incapable of turning sharp corners or passing through glass insulators. Such were the old-fashioned views upon the character and behavior of electrical discharges. As the result of carefully conducted researches upon lightning strokes and phenomena, both in this country and abroad, the old Franklin rod has undergone a radical evolution and the value of correctly designed and erected lightning conductors has been completely demonstrated. According to the latest views, lightning does not strike a direct blow to earth, possessing inertia-like qualities which prevent it from turning sharp corners. The belief in its inability to pass through glass insulators has been abandoned also, and the danger of using insulators, for other reasons, has been fully demonstrated. The design illustrated and described in the present paper was completed after a careful study of recent English and Continental systems, as well as American practice. The writer has, in addition, tested several of the latest theories by practice and experiment, and reproduces herein a few of the typical experiments with apparatus capable of giving high electrical pressures at very high frequencies. In dealing with powerful oscillatory electrical discharges many complex factors must be considered in what is generally termed the circuit. The mathematical laws governing the deportment of direct currents are wholly inadequate where lightning discharges of oscillatory character are concerned, and it has been in connection with the study of proper protection for tall power-plant chimneys that some remarkable phenomena have been noted. It is not generally known or appreciated that the column of hot smoke and gases within a tall chimney conducts as well and at times better, than the metal rods outside. This point has been taken into account in the design of the lightning equipment for the power plants of the Navy. The modern theory of lightning protection is one of quiet equalization or neutrali-

zation of unlike electrical charges, as outlined in the following extract from an earlier treatise by the present writer:

*"Conductors are looked upon by many as a source of danger, they holding that the presence of a metallic rod so prominently located enhances electrical attraction, drawing bolts from the clouds which it is seldom capable of parrying or safely conducting to earth when struck. Although the old conductor of Franklin, but partially correct in application, carefully insulated from the building to which it was attached, occasionally, or we may, perhaps, say frequently, failed to afford protection when actually struck by lightning, it may be safely asserted that thousands of buildings have been silently shielded from demolition and the dwellers unconsciously protected, through the silent equalizing action of the old rod depending upon the effect of static electrical induction. This electrical phenomenon, but imperfectly anticipated by the old inventor philosopher, yearly caused his rod to shield these buildings without receiving credit, because of the silent and, apart from the glow of blue light at the rod tips, undemonstrative action. Franklin's rod was, and is yet, carefully kept from contact with the building by mounting it upon glass insulators, overtopping gables and chimneys for the purpose of receiving a lightning stroke should an electrical cloud come dangerously near, and of offering it a metallic path to earth, which it would follow in preference to that offered by the material of the building.

"The correct and true rod should not be erected to actually receive a lightning stroke, but to prevent any flash from taking place by a quiet, equalizing action."

It appears from a careful study of lightning strokes that there are two classes of discharge, either of them being capable of demolishing a chimney if not properly provided for. The following is based upon Sir Oliver Lodge's views on electrical storms:

"It has been pointed out by Sir Oliver Lodge† that light-

*"Approved Lightning Protection." Hopkins, Nevil Monroe. Ill. "Sci. Am. Supp." No. 1212, p. 19,434, March 25, 1899.

†Lightning Research Committee report of April 10th, 1905. Great Britain.

ning discharges are of two distinct characters, which he has named the A and B flash, respectively. The A flash is of the simple type which arises when an electrically-charged cloud approaches the surface of the earth without an intermediate cloud intervening, and under these conditions the ordinary type of lightning conductor acts in two ways: first, by silent discharge; and secondly, by absorbing the energy of a disruptive discharge. In the second type, B, where another cloud intervenes between the cloud carrying the primary charge and the earth, the two clouds practically form a condenser; and when a discharge from the first takes place into the second the free charge on the earth side of the lower cloud is suddenly relieved, and the disruptive discharge from the latter to the earth takes such an erratic course *that no series of lightning conductors of the hitherto recognized type suffice to protect the building.*" The difference in character between the A flash and B flash, as named and described by Sir Oliver Lodge, may be more fully appreciated by referring to the accompanying illustrations. Here we have depicted a power plant, with its tall chimney subjected to the two kinds of lightning strain and stroke. In figure 1 at A we have before the actual stroke takes place, if it takes place at all, an inductive effect, and quiet, equalizing brush discharge, which may, if a sufficient number of sharp points are provided, actually prevent the disruptive discharge or stroke taking place. The A class may be looked upon as producing a steady strain, which may be quietly taken care of without damage to the chimney, if provided with a correctly designed and applied system of conductors. In the three following illustrations conditions are represented which give rise to the B class of strain and stroke, that which is far more difficult to safely handle by any system of conductors. In figure 2, an A type of flash between the two clouds precipitates a B type of flash between the lower cloud and the chimney top. In this case, the lightning conductors with their points upon the chimney, are not subjected to the preliminary inductive effect and strain, which may be dispelled by a silent brush discharge.

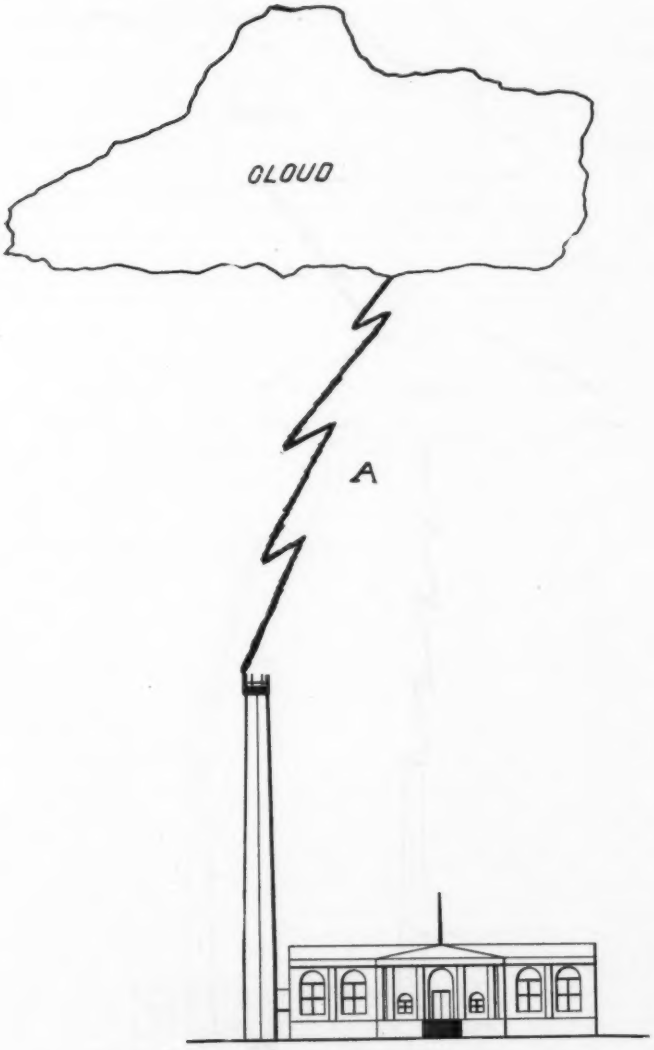


FIG. 1.

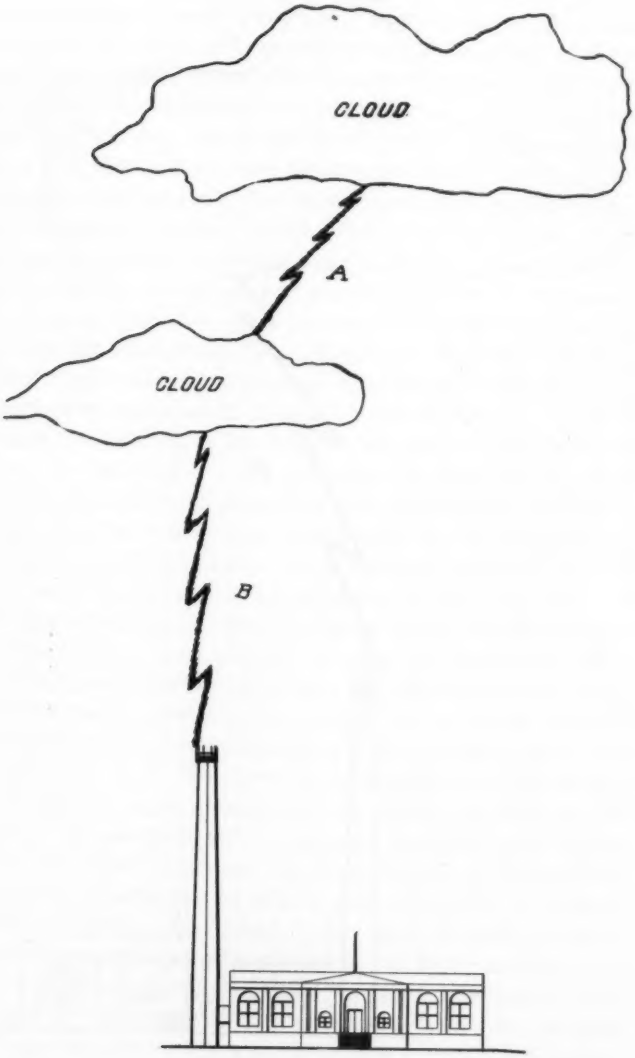
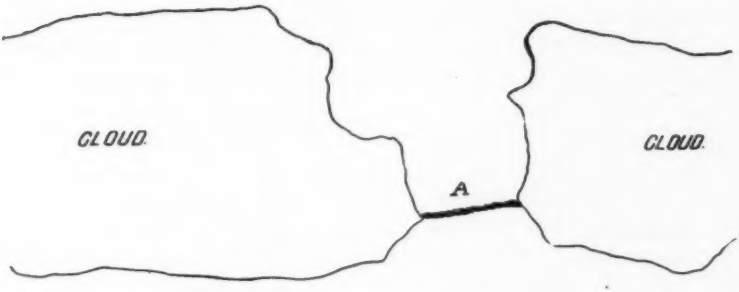


FIG. 2.



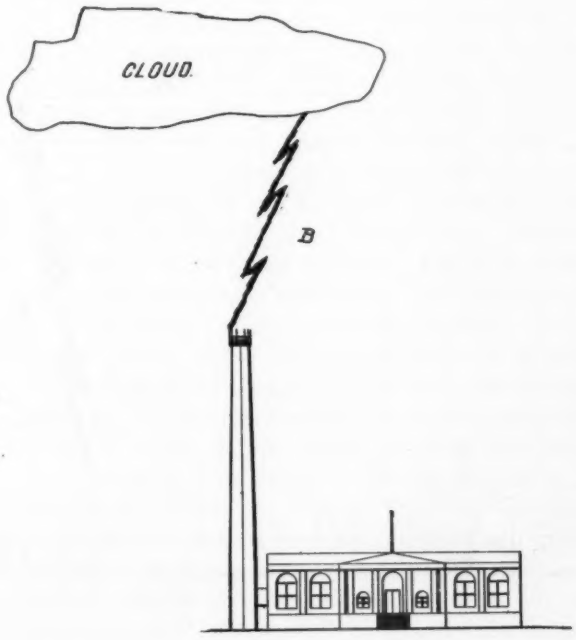


FIG. 3.

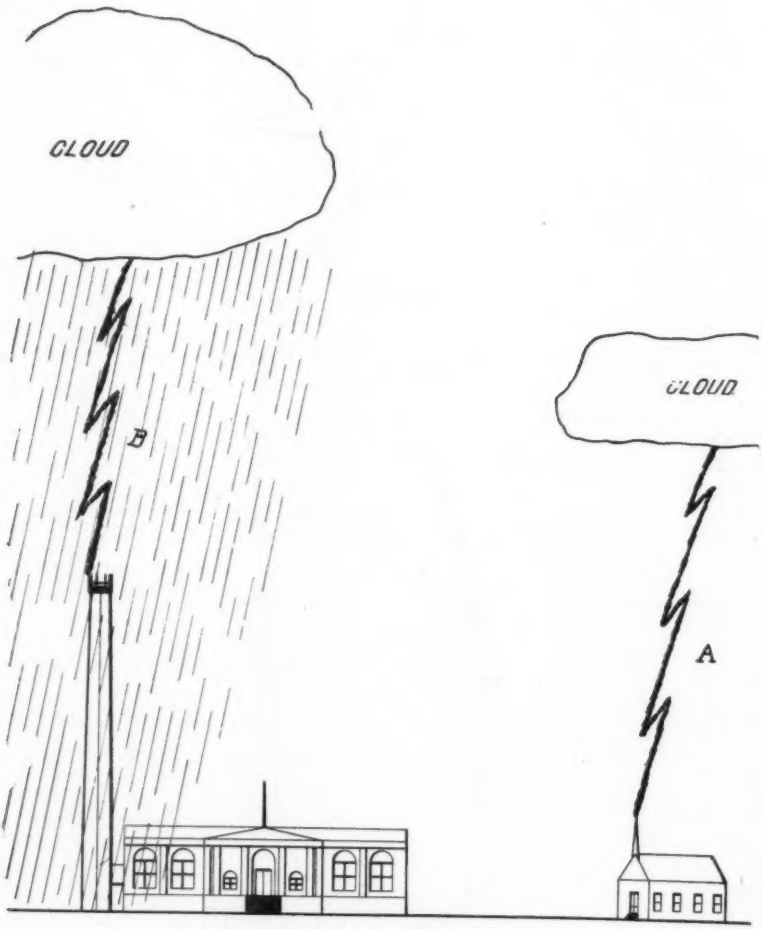


FIG. 4.

In the simple A class of discharge there is a steady, increasing electrical pressure between the cloud and the sharp point of the lightning conductors, allowing them to silently prevent a stroke.

The brush discharge, which is accompanied by the production of ozone, seems to establish a path for the discharge between the cloud and chimney, partly neutralizing its force by so doing. As already pointed out, the heated gases from a chimney act in a similar manner and will lead a discharge to a fireplace, a stove or other source of heated gases, in a dangerous manner. A properly designed and applied rod will in the great majority of cases prevent this discharge down flues and chimneys. In the case of the B class of discharge, this neutralizing action is almost wholly absent, the strain establishing itself so suddenly that an actual lightning stroke takes place at once, the lightning striking with great violence any prominent part of the building. Figures 3 and 4 show further how the A class of discharge may precipitate the B class, which takes place with such sudden violence. The only known means to protect a building against this B class of sudden disruptive discharge is by enclosing it within a wire cage, as shown by Clerk Maxwell.* This cage-like protection, although impractical to carry out fully for various reasons, has been approached in the present four-conductor rod around a chimney in the following specifications and drawings.

The present writer has demonstrated by experiment that four concentrically placed rods afford a very efficient cage-like protection to chimneys constructed on a miniature scale and subjected to the high-potential, high-frequency electrical discharge from a Tesla oscillator capable of striking through an air gap of four feet.

The experimental equipment for studying the behavior and effects of high-potential, high-frequency discharges upon a model chimney is illustrated in Figure 5. It is estimated that the model chimney and conductors are subjected to an electrical discharge at a pressure of about 1,800,000 volts, with

* "Philosophical Magazine," August, 1899.

a frequency of 200,000 oscillations per second. As will be seen, means are here provided for the establishment of an ascending current of heated gases within the chimney, by means of a Bunsen burner. It will be noted that means are also provided for breaking or opening the conductor, and also for introducing impedance in the circuit to earth, which in the present case is the other terminal of the secondary of the high-

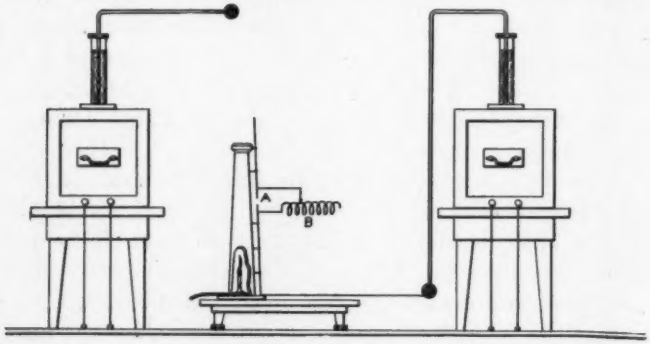


FIG. 5.

potential air core transformer designed for the very high electrical pressures and high frequencies. By lengthening the gap in the conductor at A and increasing the impedance at B, the conductivity of the conductors to these high-potential,

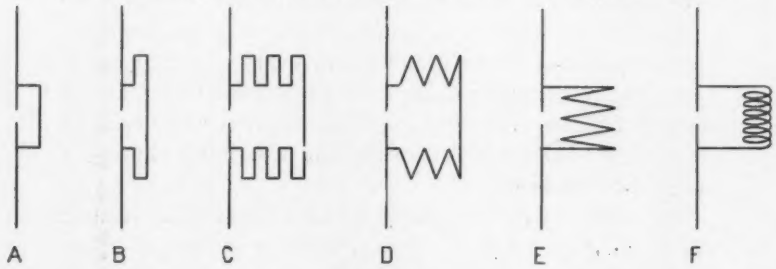


FIG. 6.

high-frequency discharges can be balanced against the conductivity of the heated column of gases within the chimney, which consist principally of carbon dioxide and water in the

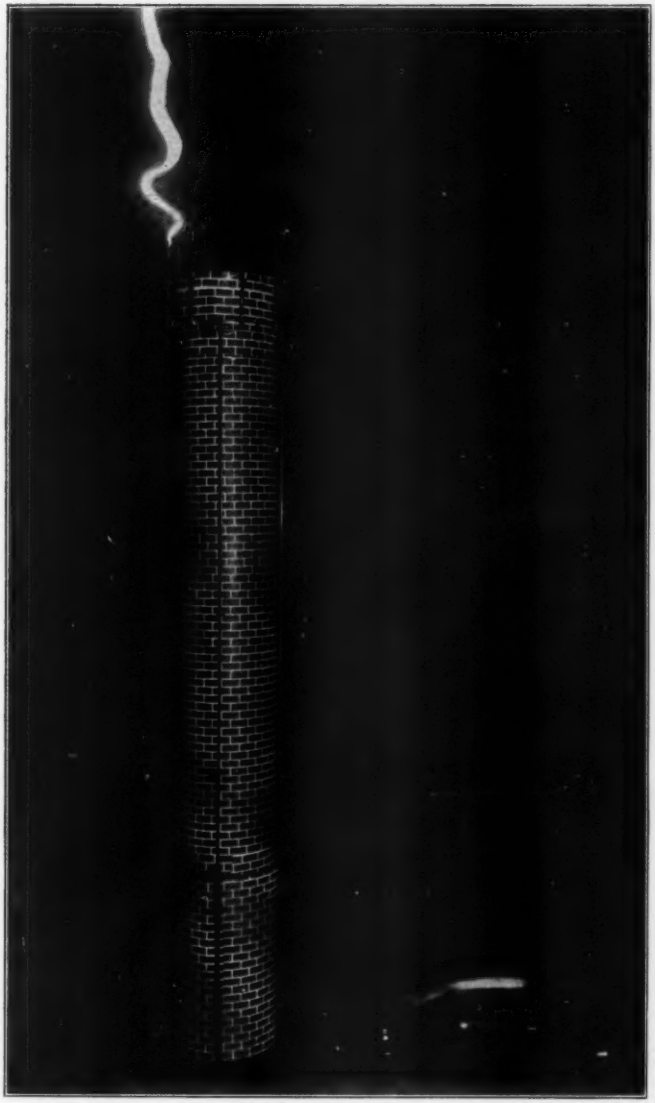


FIG. 7.

FIG. 8.

FIG. 9—A.

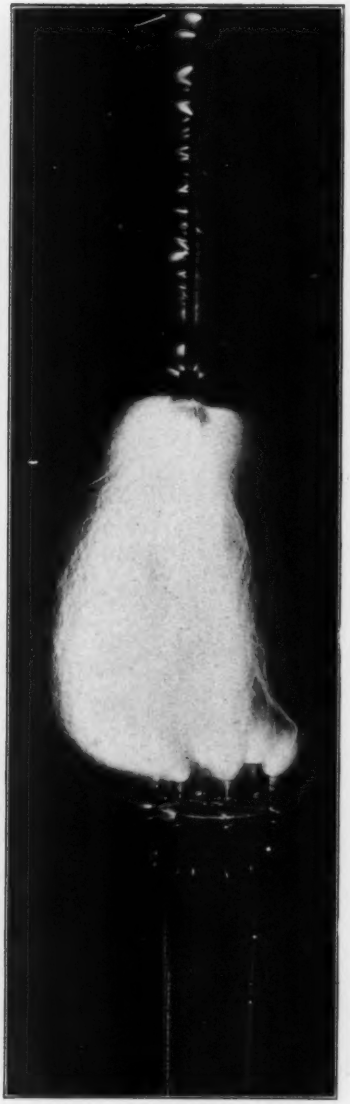


FIG. 9—B.

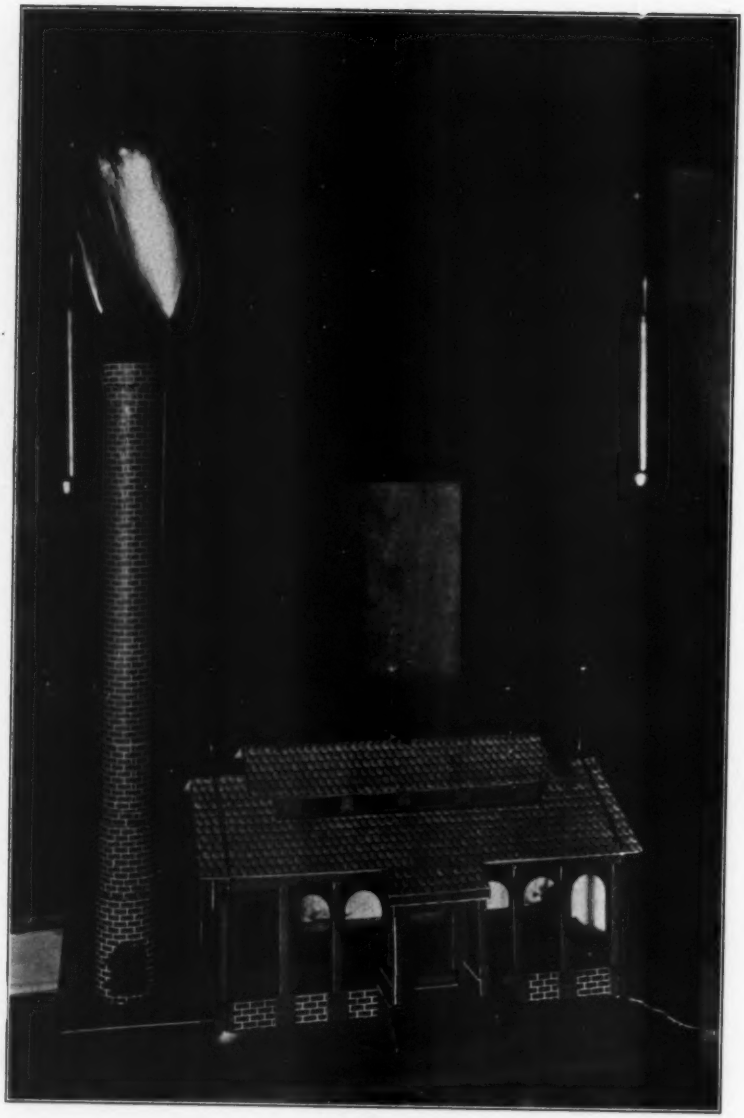


FIG. 10.

form of vapor resulting from the combustion of the gaseous fuel within the Bunsen burner.

It is easily shown that because of the great "skin effect" of high potential discharges of high frequency a discharge may descend within the chimney, tearing it to pieces from the inside, in spite of the conductors on the outside. To protect the chimneys from destruction from this character of stroke a heavy copper spider is provided as indicated in the drawings accompanying the following specifications. This spider covers the top of the chimney, the radial limbs of which run to the vertical rods. A discharge may now be directed to the ascending current of hot gases, but will strike the spider and be dissipated radially at right angles to the conductors running to earth. As it has been held for many years that lightning strokes will not readily change their directions abruptly, or pass around sharp corners, it was deemed desirable to carry

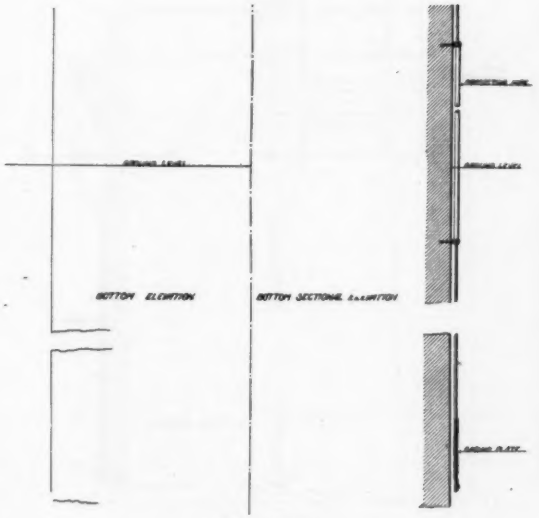
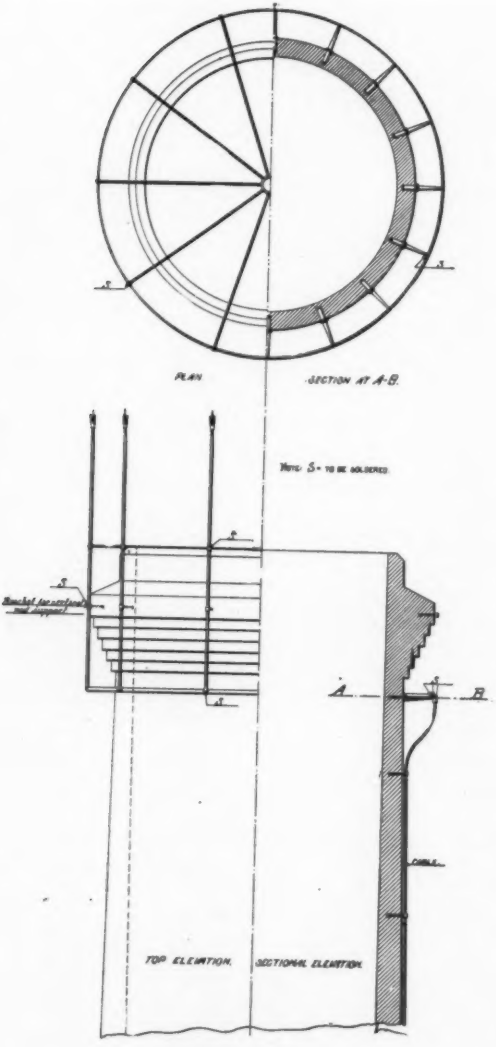


FIG. 10—A.

GENERAL ARRANGEMENT OF CHIMNEY BASE.



GENERAL ARRANGEMENT OF CHIMNEY TOP.

FIG. 11.

on some experiments to determine whether or not it was safe to eliminate from the design all easy or rounded bends.

As the installation of the spider referred to required for the protection of the interior of the chimney a right-angle deflection to the conductors to earth, experiments were made with conductors of various shapes involving sharp bends and reversals of direction. Figure 6 shows some of the forms of conductors experimented with arranged across a variable air gap. It was demonstrated that these high-frequency discharges of enormous voltages pass readily around various kinds of sharp bends and reversals in direction, as shown at a, b, c, d, e, in the figures. With any path of the nature of a spiral as shown at f, however, we of course introduce high impedance, as naturally would be expected with discharges of this kind. A full description of all experiments made, or a complete record from the writer's note book upon experimental lines would hardly be in place here, for the present

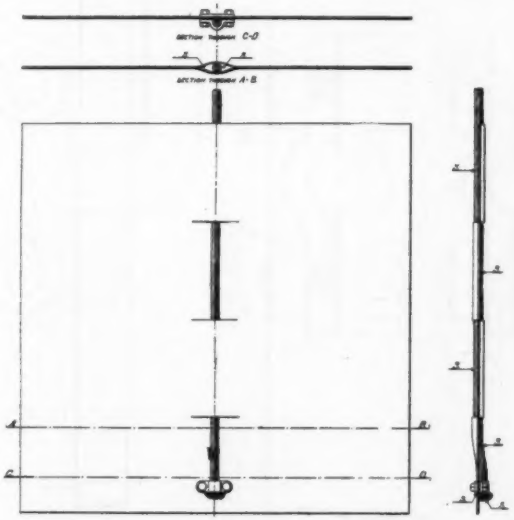
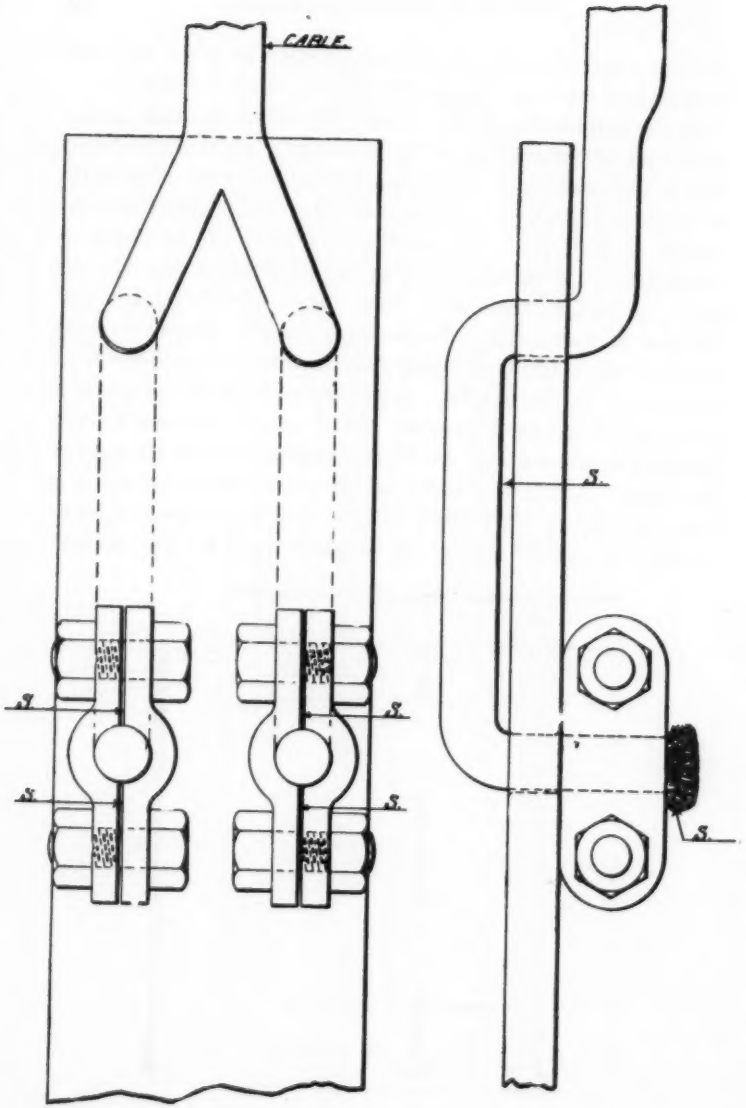


FIG. 12.

GROUND PLATE, TYPE NO. 1.

NOTE.—*S* = to be soldered.



NOTE: S - To be soldered.

FIG. 13.

GROUND PLATE, TYPE NO. 2.

FIG. 14.

SECTION OF CABLE.

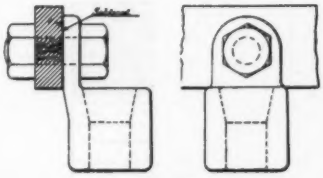


FIG. 15.

DETAIL OF CONDUCTOR ATTACHMENT TO COPPER RING

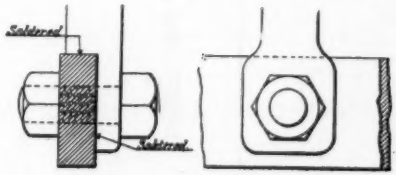


FIG. 16.

CONNECTION OF MULTIPLE TIPPED RODS AND COPPER RING.

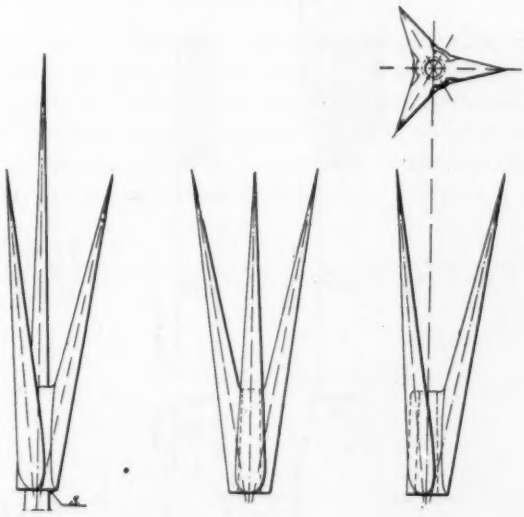


FIG. 17.

CAST POINTS.

NOTE —*S* = to be soldered.

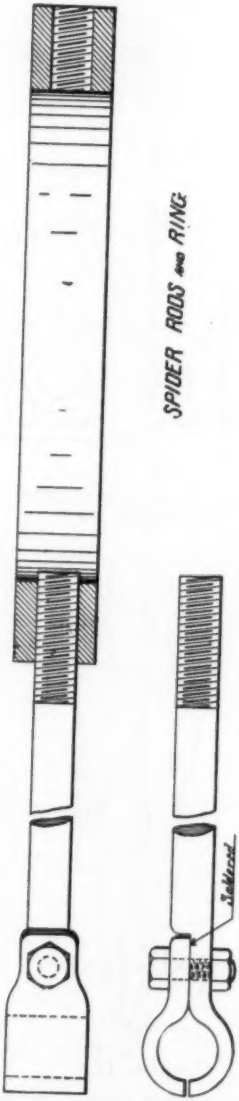


Fig. 18.

PIPE BRACKETS

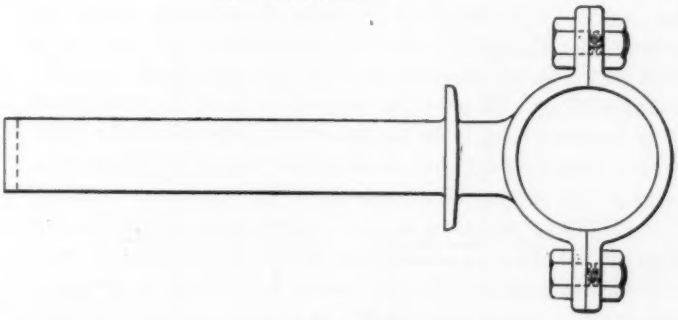


FIG. 19.

article endeavors rather to present the engineering features dealing with the practical application of conductors to tall brick chimneys. It has been the intention, therefore, to introduce only such general experiments as to show reasons for the salient features of the general design of the specifications which are given at the close of this paper. Figure 7 is from

CABLE BRACKETS.

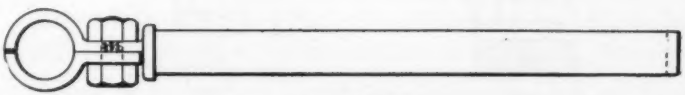


FIG. 20.

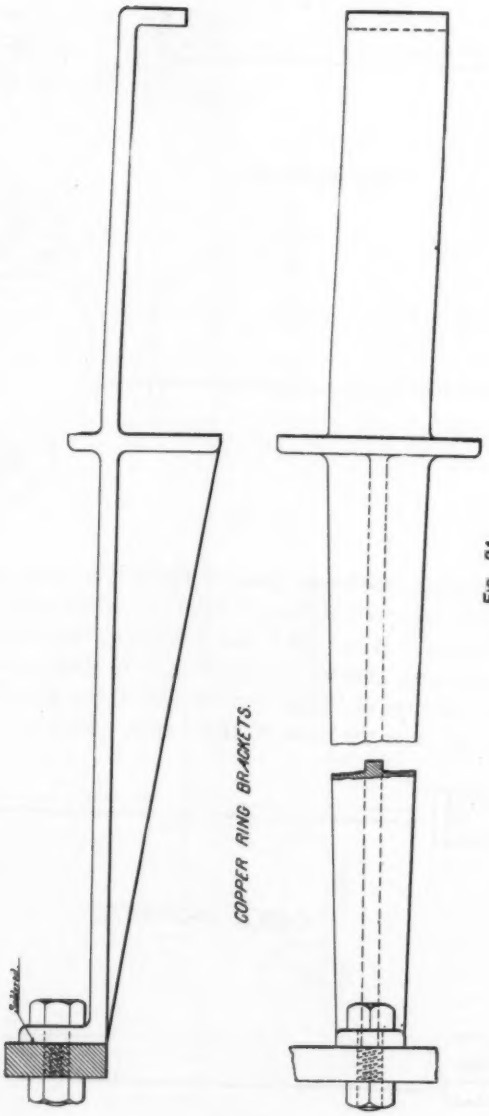


FIG. 21.

a photograph of the model pasteboard chimney experimented with. This illustrates how a B class of discharge may strike a sudden and vicious blow of oscillatory character. Here we must have ample copper area as well as conductor capacity to carry the discharge current to earth.

In this experiment the chimney top is provided with a copper spider spanning the opening, the limbs of which run radially to the four conductors to earth. The scale of the model chimney is indicated by the Bunsen burner at the right in the foreground, as well as by the ordinary 16-c.p. incandescent lamp lying alongside. The stand with glass column at the extreme right supports the impedance coil, which may be varied in inductive value and be inserted in any of the conductors to ground under experiment. In the photograph, Figure 8, we have the same chimney represented, but with the copper spider removed and exhibited at the base, the Bunsen burner being lighted within the chimney. The flash is shown striking down the interior of the chimney conducted by the carbonic acid gas and water vapor from the burning fuel in the burner. Figure 9 shows the same chimney subjected to the discharge from an electrical oscillator at close range, where the electrical strain is so great that all the eight points about the chimney top take part in dissipating the energy. This experiment represents conditions analagous to the class A discharge. Figure 10 shows a model pasteboard power house and chimney with system of conductors under the discharge of an oscillator working at diminished power but at closer range. Brush discharges are quite plainly visible from the tips of the vertical rods on the roof top. The importance of a sufficient number of sharp points surmounting all conductors cannot be overestimated.

The following is translated from the German upon the subject of providing ample streaming or equalizing points. *"The buildings of a manufacturing plant are connected by means of barbed wire strung on poles. By the connection of the barbed wire to the walls of the deeper-lying buildings, a Farraday-

* "Ueber den Blitzschutz auf Sprengstoff-Fabriken insbesondere Nitroglycerin-resp. Dynamit fabriken. Direktor Knight, Dynamitfabrik, Krümmel."

cage-like protection is secured on the one hand, and on the other hand, on account of the numerous points on the wires, a more intense equalization of the electric tension between the earth and the clouds is reached. In a dynamite factory, for example, without cartridge buildings, it is necessary to have a combination of 7 buildings with about 6,000 meters of barbed wire, 1,200,000 barbed points and 70 pole earth-conductors, and it can be readily understood that by this arrangement a stroke of lightning on such a system of barbed-wire nets dissipates its electric energy and is led to earth without leaping aside or forming secondary currents. Assembled nitroglycerine factories, dynamite magazines, and in part also the drying houses, possess this barbed-wire-netting protection, so that the aggregate of barbed wire points on the floor space of the Krümmel factory consists of more than 5,000,000."

For the complete protection of a central power plant, therefore, its roof and trusses, together with all other masses of metal without and within the building should be metallically connected with chimney conductors as well as to light rods running along the top of all roofing and other prominent parts of the building. Sharp points should be placed at close intervals somewhat analogous to the protection afforded by the barbed-wire lines as described in the German article. As the architecture of the building must necessarily dictate the precise arrangement of conductors, the specifications as herein given pertain only to the protection of the chimney which, if properly provided for, because of its towering height, affords also good protection for the building. The present paper closes with the following specifications as approved for the protection of the brick power-plant chimneys of the Navy.

STANDARD SPECIFICATION FOR LIGHTNING RODS.

Chimney Protection for Power Plants.—Lightning conductors shall be laid up in the form of a seven-strand cable and each strand laid up with seven copper wires of No. 10 B. & S. gauge. For chimneys of 50 feet and less in height, two lightning conductors shall be used. For chimneys over 50 feet up

to and including 100 feet, three conductors shall be installed. For chimneys higher than 100 feet, four conductors shall be nstalled. All heights to be considered from ground level. All conductors or cables shall be symmetrically arranged about the chimney with one cable on the prevailing-weather side of the chimney. Said lightning conductors or cables to be securely attached both mechanically and electrically to independent pure copper earth plates or bars. In cases where the chimney foundations have already been filled in, instead of earth plates earth terminals may be used, composed of pure copper bars 3 inches by ½ inch by 3 feet. In all cases the lightning-conductor terminals shall extend to the ground water level, and in no case shall they extend to less than 15 feet from the ground surface. Earth plates shall consist of pure copper 3 feet by 3 feet by ⅛ inch.

Application of Conductors to Chimney.—Each lightning conductor shall be secured to the exterior of the chimney by means of bronze or brass anchors, without the intervention of any insulators or insulating material whatever. The brackets for attaching ring or conductors to chimneys to be of high-grade bronze or brass, composition of same to be submitted for approval, and to be fitted with approved clamps for securely gripping said conductors and making a good electrical connection therewith. The tongues or shanks of the anchors or brackets shall enter the masonry of the chimney a distance of at least 6 inches, and shall be at least ⅛ inch in thickness by 1 inch wide, terminating in a suitable head or angle, to prevent the anchor from being pulled out of the masonry. Anchors to be attached to conductors at intervals of not over 10 feet, and sweated to the conductors with solder at intervals of 50 feet. Conductors to terminate within 5 feet of the top of the chimney, and to be connected through the agency of suitable brass or bronze fitting and be soldered to a 1½-inch by ½-inch ring of copper attached to the periphery of the chimney by brackets spaced not over 2 feet apart; said brackets to enter the brickwork a distance of at least six inches and to be of approved design, with a tongue at least 1½ inches in width and

$\frac{1}{4}$ inch in thickness, with a suitable angle or head to prevent pulling out. All joints in the continuity of said copper ring as well as between the continuity of the ring and conductor or conductors running down to the ground bars or plates, and including the latter, to be scraped bright and, after making a secure mechanical joint, to be "sweated with solder." Said solder shall consist of $\frac{1}{2}$ lead and $\frac{1}{2}$ tin. All joints when finished shall be thoroughly washed off with water to remove every trace of soldering salts, acids or other compounds used. All joints secured by bolts or screws to be locknutted. In applying conductors where the chimney is already constructed holes shall be drilled in the brickwork and said anchor brackets and anchors grouted in, the best Portland cement being used.

Terminal Rods for Lightning Conductors.—Copper ring shall be connected through the agency of clamps, insuring a good mechanical and electrical joint, with vertically-arranged copper rods, at least $\frac{3}{4}$ inch diameter and 10 feet in length. The joints to be "sweated with solder" as before described. Copper rods to be placed equidistant around this ring, and supported in a rigid position vertically through the agency of additional anchors set in the masonry and a copper spider resting on chimney top as shown in the drawings. Rods to be arranged with a uniform spacing of practically 4 feet. This is taken to mean, for example, that ten such vertical rods shall be provided for a chimney of 12 feet outside diameter of chimney at top.

Discharge Points.—Each rod shall terminate in a 2-point aigrette, each spur or point of this aigrette to be at least $3\frac{3}{4}$ inches long, the bases of which spurs shall be at least $\frac{3}{8}$ inch in diameter, tapering to a sharp and well finished point; said aigrette to be provided with approved means to secure a strong mechanical and electrical joint with the vertical rods heretofore described and to which it is attached. The joints shall be "sweated with solder" as heretofore described.

Chimney-Base Protection.—All lightning conductors shall be enclosed at bottom with a heavy galvanized iron pipe of $1\frac{1}{2}$ inches diameter, and extending 3 feet into the soil and 10

feet above. Said iron pipe to be provided with approved brackets to securely hold it to the chimney; brackets to be not over 3 feet apart.

Inspection.—No scaffolding shall be removed before inspection of the lightning-conductor installation shall have been made by a Government representative.

Note.—This specification is to be accompanied by a drawing, furnished by the Bureau of Yards and Docks and approved by the same, giving detail parts.

TORSIOMETERS.

TORSIOMETERS AS APPLIED TO THE MEASUREMENT OF POWER IN TURBINES AND RECIPROCATING ENGINES.*

By ARCHIBALD DENNY.

When the suitability of the turbine method of propulsion for commercial work was proved by the success of the *King Edward*, built by my firm in 1901, it became apparent to us that it would be highly desirable to have a method of ascertaining the horsepower transmitted by the turbine shafts to the propellers. Until that problem was solved we could only work from the boiler to the propeller, and the efficiency of the turbine and the propeller must be lumped together. It is not possible to "indicate" the turbine in the same way as is done for a piston engine, although I may say that a fair approximation can be got by ascertaining the fall of pressure through successive expansions by means of pressure gauges fixed to the turbine casing.

Some fifteen years ago we had made numerous experiments with factory shafting, endeavoring to ascertain the absolute torsion of a shaft while running, and it therefore immediately occurred to me that this was the proper direction in which to attack the problem. We had tried various methods, principally using pierced discs and beams of light, but with very partial success. We had not tried any method involving the use of electricity, and I therefore arranged for experiments to be made by this method on one of our factory shafts. The first trials were made by fixing discs on the shaft at a considerable distance apart, so as to get a reasonable amount of torque. The discs were of insulating material, and each had

* Paper read before the Institution of Naval Architects, March 21, 1907.

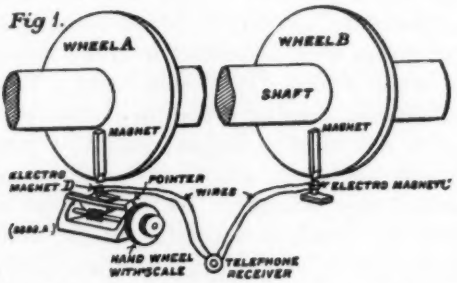
a contact point arranged at its periphery in such a manner that the point made momentary contact with a metal tongue or brush once in every revolution of the shaft. The contact points were connected to the shaft, and the metal brushes to a battery and a telephone receiver. The method adopted was to first adjust the brushes so that both made contact with the points simultaneously when the shaft was revolving, but transmitting no power. When transmitting power the shaft was, of course, subject to a certain amount of torsion, and thus the brushes were put out of simultaneous contact. One of the brushes was then moved round its disc concentrically, until simultaneous contact was once more established. The amount of this shift gave a measure of the torque on the shaft, and to ascertain the correct amount of this shift the telephone receiver was placed to the ear, no sound being heard except when both brushes were in contact with the respsctive contact points, when a loud "tick" was heard. The principle was thus of extreme simplicity, and the method of carrying it out seemed at first equally simple; indeed, I may say that this first rough apparatus, which was quite successful, cost only a few shillings to make. We then set about making more accurate and elaborate apparatus on the same lines, to be fitted to the *Queen Alexandra*, which was nearly ready for trial, with an assured hope of getting satisfactory results.

The factory shaft on which we made the original experiments ran about 120 revolutions per minute, but the revolutions of the *Queen Alexandra's* side shafts were over 700, and when we came to make experiments at this high speed we found the new apparatus was useless, as no certain sound could be got. We tried many forms of contacts, and after numerous experiments we did succeed in the *Queen Alexandra*, with revolutions about 750, in getting some fairly consistent results; but it was impossible to be quite certain of the exact point at which the make-and-break in the circuit took place, and we were never quite sure of our results; still we had made a great step in advance.

Mr. Charles Johnson, a member of our staff, who assisted

in working out this problem, and was closely connected with it from the first, thoroughly appreciated the difficulties, and realized the desirability of getting away from the unreliable rubbing contact, and he ultimately succeeded in solving the problem in a most ingenious way.

Fig. 1 and Fig. 8 show his original solution. Two gun-metal wheels, A and B, were fastened to the shaft at a definite and known distance apart, the distance being as great as possible. On each wheel a permanent magnet, with a sharp, chisel-shaped edge, was fixed radially at the periphery of the wheel and with the sharp edge parallel to the shaft. At one end a soft-iron electro magnet C, wound with fine wire, similarly chisel-shaped, was fixed, so that the moving magnet



passed directly over the electro magnet once in each revolution. At the other end a similar electro magnet D was mounted on a screwed sector, and wires from these electro magnets were led to a differentially-wound telephone receiver. If the shaft revolved without transmitting power, the permanent magnets passed the electro magnets simultaneously, and currents of electricity generated in each coil passed through the telephone receiver; but the currents being equal and opposite, no sound was heard. When the shaft transmitted power the permanent magnets passed the electro magnets at different times, and hence a sound was heard in the receiver. By turning the hand wheel shown in the diagram a new position of silence could be obtained, when it was evident that the two permanent magnets were again passing the electro mag-

nets simultaneously, and the amount of torque could be ascertained from the reading of the sector screw.

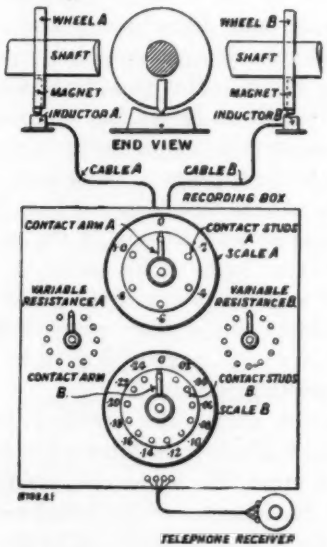
This instrument was used with great success on a number of the turbine vessels built by my firm, but it required the operator to work in the tunnel, where, even with turbines, there is some noise and disturbance; and to overcome this objection Mr. Johnson designed another arrangement, which is illustrated in Fig 2. The gun-metal wheels and permanent magnets remain as before, but in place of the chisel-shaped electro magnets, soft-iron sector cores, marked inductors A and B, take their place. These cores are wound with a series of coils of very thin wire, so fine, indeed, that each coil, with its dividing wall in one of the sectors, only occupies a space of 0.02 inch. This sector, or indicator, as we term it, is fitted underneath one of the gun-metal wheels, and a similar one is fitted at the other wheel, but in this case the coils are further apart—namely, 0.2 inch; the reason for this will be afterwards explained.

In Fig. 2 is also shown the recording box, which will be seen to consist of two circles fitted with contact studs and movable contact arms; the upper one, marked "Scale A," is connected with the inductor at the wheel next the propeller, and has six contacts; that is to say, there are six coils, spaced 0.2 inch apart. The other one—"Scale B"—has thirteen contact studs, which are similarly connected to thirteen coils on the inductor at the turbine end of the shaft; these coils are spaced 0.02 inch apart. On the recording box will be seen two side coils, marked respectively "Resistance A" and "Resistance B." These are resistance coils for throwing into series with the differential windings of the telephone receiver, which is connected to the two inductor circuits, which circuits must be accurately balanced before absolute silence can be got in the receiver.

Fig. 2 shows the arrangement of the apparatus for one shaft. With this arrangement it is possible to locate the observer in any part of the ship, cables being led from the inductors in the tunnel to the recording box, and we usually

select some quiet cabin where the observer will be undisturbed. The sound in the receiver at usual revolutions is so distinct that even an untrained observer, after a very few min-

Fig 2.



utes' practice, can get perfectly accurate results, and powers transmitted can be ascertained from moment to moment.

It is essential, of course, that the correct zero position, from which to start, should be found. In order to do this, zero lines, or, rather, grooves, are cut into the cover plate of the inductors A and B, directly over the zero coils; and in setting up the instrument the chisel-shaped magnets are brought directly over these lines, by first dropping them exactly into the grooves, and then raising and fixing them to just clear the top of the inductors. But we do not depend upon this method alone, because any inaccuracy in setting up the inductors might bring in small errors. We therefore get an absolute zero by running each shaft light alternately; that is to say, before the commencement or after the completion of

the trials, each shaft is alternately run light, the vessel being driven by the other two screws. It is true that by this means we do not eliminate the effect of friction in the turbine and shaft bearings; but this must be very small, as we find our original method of setting the zero, when carefully done, is always confirmed by this method of checking the same.

To take a reading after the instrument is once set, and the resistance of the two telephone circuits is adjusted by means of the variable resistances A and B, all that is necessary is to turn the movable arm on scale B round the various studs until there is silence in the telephone, when the amount of torsion is immediately read off the scale. If no such position be found, it means that the shaft is being twisted more than is covered by this scale; the arm A is then turned to the first contact, and the arm B is again swept round the circuits. If silence be still not obtained, the arm A is turned to the second contact, and so on, the combined range of the scales being altogether 1.24 inches, which is more than sufficient to measure the maximum torque usually obtained. From torsion experiments on the shaft described below, made previous to its being fitted in the ship, or from a formula also given below, the factor by which this reading is to be multiplied is obtained, and the power is got by a simple multiplication sum.

This instrument is made for one, two, three and four shafts; for each shaft a group of circles as shown is required. By means of the contact arms and studs the various shafts are thrown into circuit with their receivers, and readings are taken from three or four shafts in a very short time. This instrument is not suitable for very low revolutions, as the induced current becomes too weak to make a distinct sound in the telephone; but it is suitable down to about 100 revolutions per minute. For lower revolutions the instrument first described is employed, but the operator must then be in the tunnel.

The resting of the shaft to get the torsion scale previous to its being fitted on board is done by fixing rigidly one end of

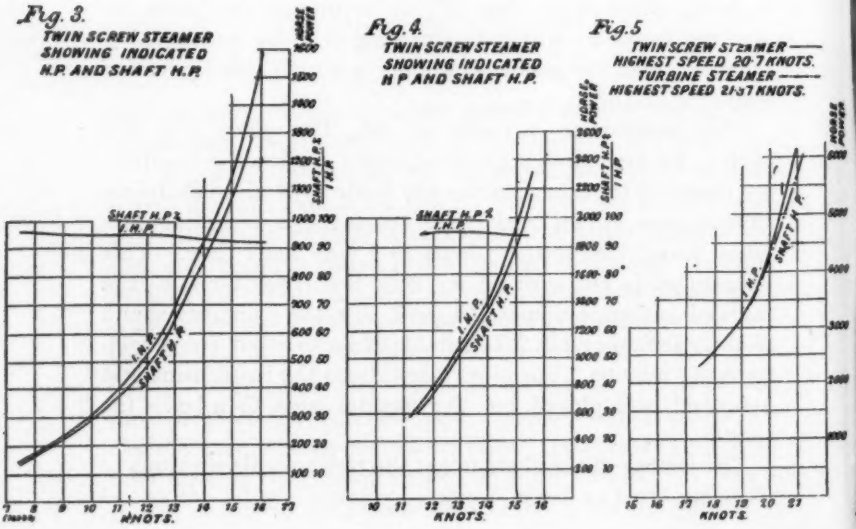
the length of the shaft to be used for the trials, and at the other end fitting a lever; this is loaded with weights, so as to get the scale of torsion moment. For turbine shafts, which are small in diameter as a rule, this is not a serious operation, nor, indeed, have we found it so even for larger ordinary twin-screw shafts, although, of course, the weights used have to be much greater. We have tested all the shafts for the turbines and other vessels on which we have used these instruments, and we find that the torsion is given very closely by the following formula:

$$\theta^0 = \frac{WRL}{Kd^4},$$

where

θ^0 = angle of torsion in degrees.
WR = foot-pounds turning moment.
L = length of shaft in feet.
d = diameter of shaft in inches.
K = coefficient.

K may be taken at 140 for mild-steel shafts where there are no couplings, or, if there be any, by deducting their length



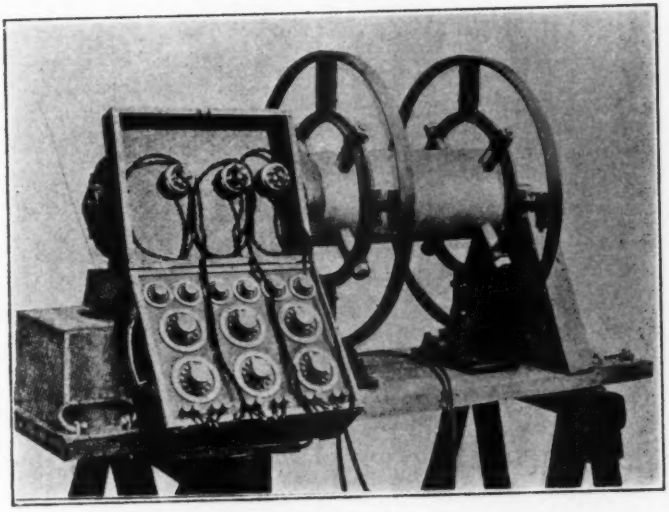


Fig. 6.

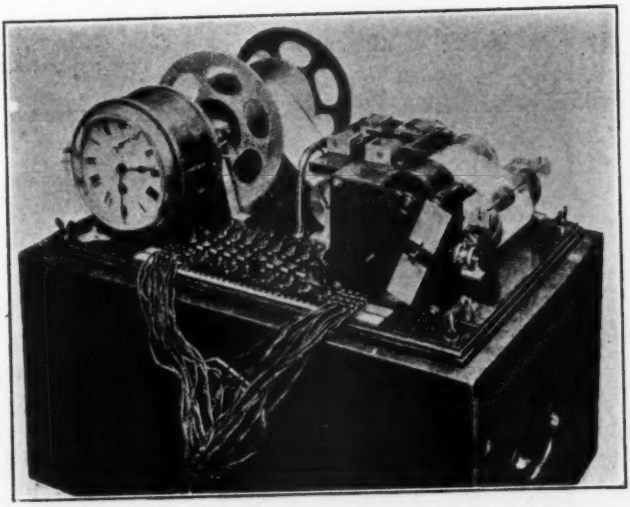


Fig. 7.

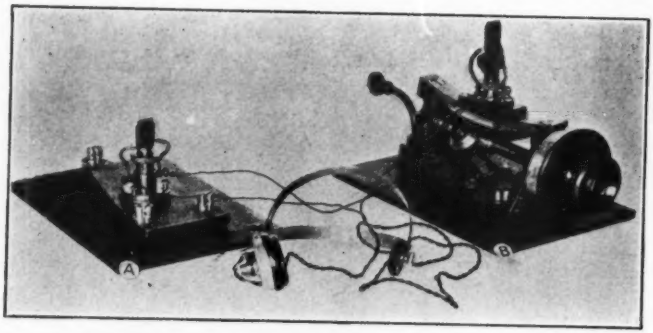


Fig. 8.

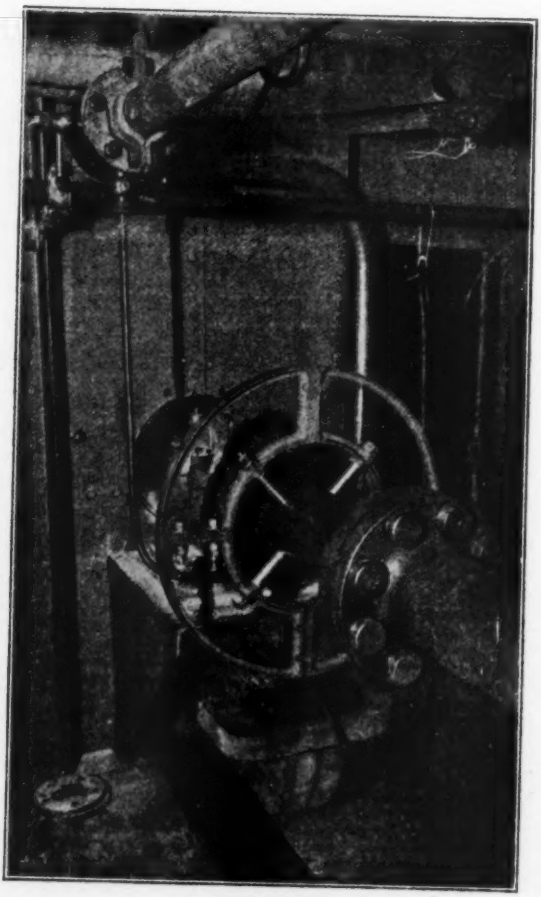


Fig. 9.

from the length of shaft twisted—that is, we assume that the couplings do not twist. It is, of course, better to test each shaft independently, but the error would not be great by using the formula—probably not 1 per cent. either way.

While this instrument was designed primarily for turbine engines, it was quite natural that we should attempt to use it for ordinary piston engines as well. In that case, however, on account of the variable torque throughout a single revolution, not less than six magnets ought to be fitted round the gun-metal wheel, and six sets of inductor sectors. This has actually been done in the case of two small twin-screw vessels with success, but it was a very cumbersome arrangement.

The diagrams in Figs. 3 and 4, give the results of the experiments, from which it will be seen that they are very consistent, and show about 94 per cent. efficiency for the steam engine.

Fig. 6 shows a set of recording instruments for three shafts, and the wheels and sectors for one shaft fitted to a dummy; the actual instruments are exhibited on the table. Fig. 9 shows one of the wheels as actually fitted in a tunnel.

As a still further convenience, we use on all our trials of turbine vessels another machine, shown in Fig. 7, with a roll of paper, on which a time pen works, and also pens recording the revolutions of the turbines; and for convenience, these are only recorded for every tenth revolution of the turbine. This machine is also connected by electric cables with the bridge, by means of which the beginning and the end of the mile can be signalled, and thus we are able, at one recording station, to ascertain the mean speed, the mean revolutions and the mean power, all taken by one or two observers in a quiet and undisturbed cabin.

The diagram in Fig. 5 shows the results got from two practically sister vessels, one fitted with twin screws, and the other with turbines.

The great advantage of these instruments is that we separate and define the efficiency of the turbine and the screw,

and by this means we have been able to discover which is the best combination for any particular ship.

I feel we owe a great deal to the skill and perseverance of Mr. Charles Johnson, a son of the late Mr. Charles Johnson, a member of this society, who was with us for many years, and latterly in partnership with the late Mr. William John.

TORQUE OF PROPELLER SHAFTING.

TORQUE OF PROPELLER SHAFTING: SOME INVESTIGATIONS AND RESULTS.*

By J. HAMILTON GIBSON.

The necessity of ascertaining the horsepower of the marine steam turbine by some means analogous to the "indicator" has revived interest in the torsion meter as applied to propeller shafting.

Indicated horsepower, as obtained by mean-pressure diagrams from a reciprocating engine, is all very well from the naval architect's point of view so long as frictional and other losses are known and remain fairly constant; but the actual power transmitted to the propeller by the shaft is more definitely valuable information, if such is obtainable and can be relied upon.

In utilizing a steel bar, such as a propeller shaft, for the measurement of power, obviously the first thing to do is to "calibrate" that portion of the shafting which is to be so utilized; for the shaft behaves exactly like a steel spring. However accurate the dimensions and homogeneous the material, there are slight differences from the calculated results that must be known; just as there are discrepancies in a set of apparently identical helical springs for boiler safety valves. Each must be tested and set independently; and the analogy suggests that shafts used for torsion-meter records should be recalibrated periodically, just as safety-valve springs are.

A screw-propeller shaft, when transmitting power, undergoes a compound strain. Not only is it twisted by the engine; it is simultaneously compressed by the thrust of the propeller. That this thrust is not an entirely negligible quantity may be inferred from the fact that in an actual case, the torsional

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* Paper read before the Institution of Naval Architects, March 21, 1907.

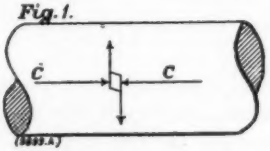
stress in the shafting at full power being 5,100 pounds, the compressive stress due to thrust works out at 950 pounds per square inch, or 18 per cent. of that due to torque alone.

In calibrating the shafting, then, it is advisable to make an allowance for, or to imitate as far as practicable, the actual working conditions, and to fix the power constant accordingly.

Appendix No. I shows by calculation the probable effect of the compound stress referred to; but it is given with all reserve, as actual experiments do not show quite so marked an effect. This is probably due to the fact that, even when transmitting full power, a propeller shaft is rarely twisted more than 1 degree in 10 feet, and the effect of compression will naturally be more and more marked as the twist increases.

Thus a minute square on the surface of a shaft will become skewed and form a rhombus as the shaft twists, and the square will be pushed out of shape more easily by axial compression C, C, Fig. 1, as the shaft goes on twisting. It follows, also,

Fig. 1.



that the effect of compression will be more marked in a hollow shaft than in a solid shaft, and in a shaft having a very large hole than in one of smaller bore; for the skewing effect becomes greater as the surface of the shaft is approached, whilst the core remains comparatively unaffected by the torque, and offers an unimpaired resistance to compression.

Figs. 2, 3, 4 and 5 are descriptive of an actual calibration experiment carried out in accordance with these ideas.

The torque and compression were calculated beforehand for the varying speeds, allowing 95 per cent. of the estimated equivalent indicated horsepower for torque, and assuming 65 per cent. of the same to be re-transmitted from the propeller to the vessel in the form of thrust. In other words, the turbine efficiency was taken as 95 per cent., and the propulsive

efficiency as 65 per cent., giving a propeller loss of 30 per cent. The figures are necessarily somewhat arbitrary, but sufficiently near for the purpose.

Two light pointers were secured to the shaft at a certain fixed distance apart, and of such a length that their points

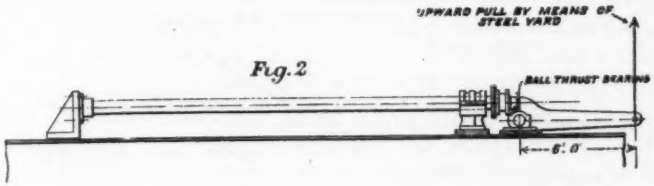


Fig. 2

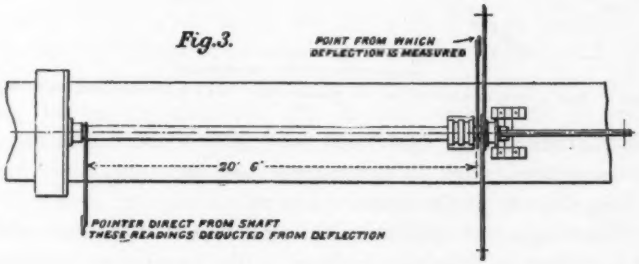


Fig. 3.

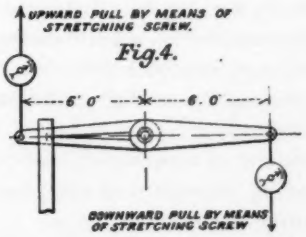
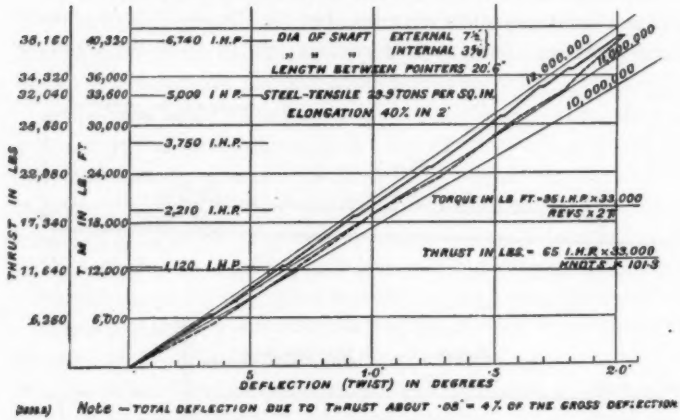


Fig. 4.

described 1 inch of arc for 1 degree of torque. The torque could thus be readily measured by a fitter's decimally-divided straight edge. The difference between the pointer readings gave the exact torque. It is worth notice that, however secure the fast end of the shaft may be made, there is always

Fig. 5 CALIBRATION TEST OF STEEL SHAFTING.

COMBINED TWISTING MOMENT AND CORRESPONDING THRUST T.M. APPLIED FIRST, THEN BEING INCREASED CONTINUOUSLY. RETURN LINE (DOTTED) GIVES THE EFFECT OF DECREASING THE LOADS (T.M AND THRUST) SIMULTANEOUSLY, AND AT THE SAME STAGES AS ON THE UPWARD LINE



Note.—TOTAL DEFLECTION DUE TO THRUST ABOUT ·08" = 4% OF THE GROSS DEFLECTION

the liability of some movement, especially at the beginning of an experiment, due to fastenings, such as bolts, keys, &c., giving slightly as the stress comes on.

The torque was applied by spring balances at opposite ends of a couple, to eliminate friction in the supporting bearing, whilst the shaft was compressed by means of a 6 to 1 bell-crank lever loaded by weights on a steelyard. To enable the end of the shaft to turn freely, a ball bearing was introduced between it and the compression lever.

Referring to the diagram in Fig. 5, the loads were applied in the following order:

1. First increment of torque.
2. Corresponding increment of compression.
3. Adjust torque balances.
4. Second increment of torque.
5. Second increment of compression.
6. Adjust torque balances. And so on.

An appreciable time was allowed between each increment for the shaft to accommodate itself to the stress. The loads were taken off in inverse order.

The effect of adding the compression each time was not to increase the angle of torque sensibly and directly, as might be expected, but to slightly relieve the loads on the torque balances. When, however, the torque balances were screwed up again and adjusted to their former load, the shaft was found to be twisted an additional amount, represented by the short horizontal lines on the upward curve, and these horizontal lines became appreciably longer as the full load was attained. The phenomenon of mechanical hysteresis or "lag" was very marked on the return curve, shown dotted; but on the removal of the last increment of load the pointer immediately returned to zero. As a check, the experiment was repeated several times with variations, such as applying the loads quickly, and taking off the total load suddenly; but the curves followed the same lines within narrow limits, and the shaft invariably and instantly returned to zero, thus proving its perfect elasticity, but leaving some doubt in one's mind as to the real nature of the "lag" referred to. Apparently a higher modulus should be taken for a shaft in which the power is gradually and uniformly augmented, and a lower modulus as the power is gradually and uniformly reduced. But these conditions do not obtain in actual work. A constant fluctuation of load giving rise to torsional oscillations in the shaft occurs, which probably has the effect of harmonizing the maximum and minimum torque for any given power; and the true modulus might be taken as a mean between the upward and downward curves. In the present case the average modulus is 11,250,000, giving a constant of 3.27 in the horsepower formula.

Thus

$$H = \frac{\theta D^4 N}{C L} \text{ for solid shafting,}$$

and

$$H = \frac{\theta (D^4 - d^4) N}{C L} \text{ for hollow shafting,}$$

where

H = shaft horsepower.

θ = angle of torsion in degree.

D = diameter of shaft in inches.

d = diameter of hole in shaft in inches.

N = number of revolutions per minute.

C = 3.27 (corresponding to a modulus of rigidity of 11,250,000)

L = length of shaft in inches.

In a more recent experiment, the effect of keeping the shaft "alive" during the calibration test was tried with good results. Two lengths of 6½-inch solid-steel shafting were coupled together, as shown in Fig. 6, and were kept "on the dither" by tapping the surface with lead hammers.

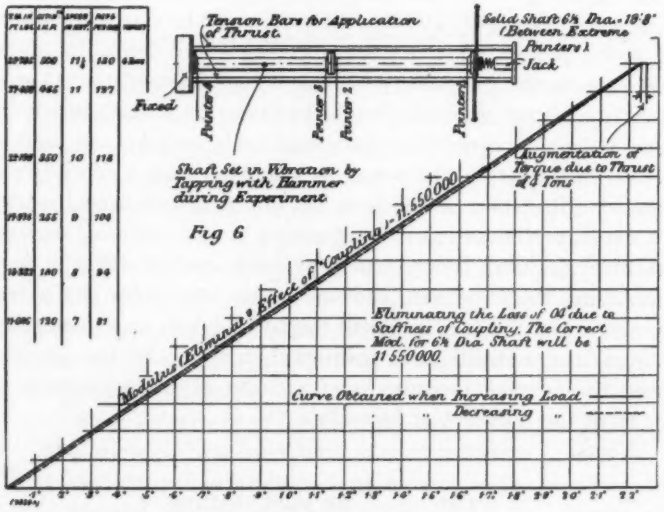

Tension Bars for Application of Thrust.
Solid Shaft 6½ Dia. - 15'8" Between Extreme Pointers
Jack
Fixed
Pointer 4
Pointer 3
Pointer 2
Pointer 1
Shaft Set in Vibration by Tapping with Hammer during Experiment
Augmentation of Torque due to Thrust of 5 Tons
Modulus Eliminating Effect of Coupling = 11,550,000
Eliminating the Loss of Oil due to Softness of Coupling. The Correct Mod. for 6½ Dia. Shaft will be 11,550,000
Curve Obtained when Increasing Load
" Decreasing "

Fig 6

It will be noticed that the lag in this case is almost imperceptible, and that the curve follows a straight line, as, indeed, it should do within the limits of elasticity. Evidently it is important to set up and maintain a species of molecular vibration in making such tests as we are considering. Two

additional pointers were used in this experiment to measure and eliminate the effect of the center coupling, and the compression corresponding to the propeller thrust at full power was added by a screw jack through a calibrated spring and ball bearing after the final increment of torque. The augmentation of torque recorded was just 1 per cent., the torque balances going back ¼ cwt. from 25 cwt. The horizontal line at the top end of the curve shows the additional torque on screwing up the balances again to 25 cwt. The experiments, so far, appear to show that an augmentation of torque due to propeller thrust of from 1 to 1½ per cent. for solid shafting, and from 3 to 4 per cent. for hollow shafting, is a sufficient allowance to make under ordinary working conditions for shafts of normal proportions.

Fig. 7 WING SHAFT 7¼ DIA LENGTH 18' 7½"
MODULUS 12"

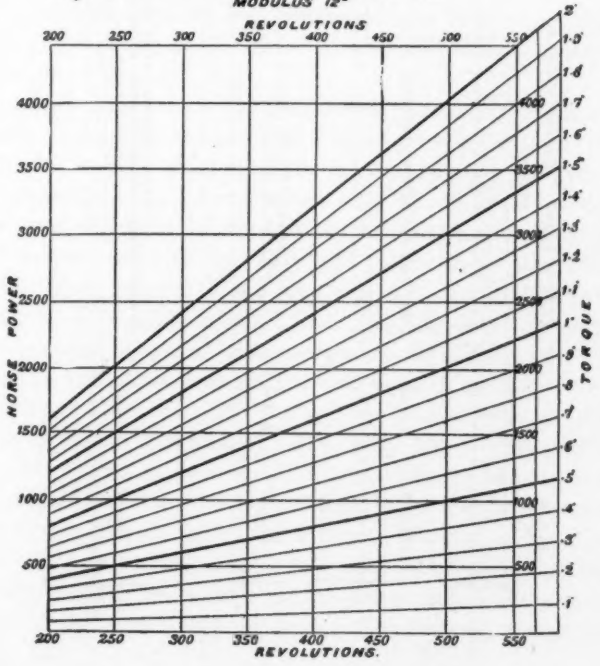


Fig.8. FLASH LIGHT TORSION METER.

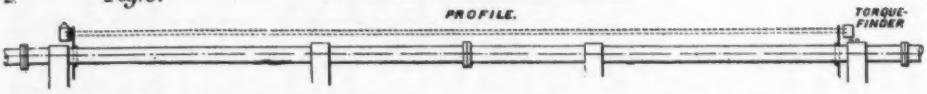


Fig.9. PLAN. NO TORQUE, (LIGHT VISIBLE)

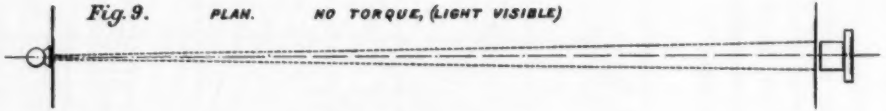
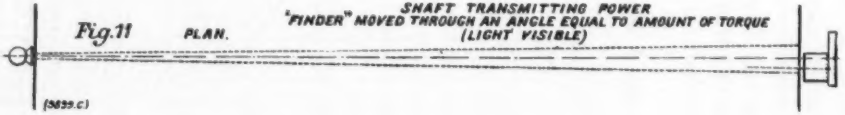


Fig.10 PLAN SHAFT TRANSMITTING POWER. (LIGHT OBSCURED)

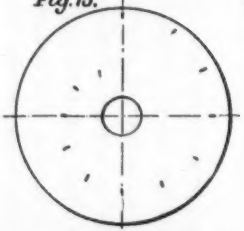
Fig.11 PLAN. SHAFT TRANSMITTING POWER "FINDER" MOVED THROUGH AN ANGLE EQUAL TO AMOUNT OF TORQUE (LIGHT VISIBLE)



(5839.c)

Fig.12.

Fig.13.



DISCS PERFORATED THUS FOR RECIPROCATING ENGINES

The horsepower formula is such that for a shaft of given diameter, length and ascertained elasticity a diagram as shown in Fig. 7, can be readily constructed to enable the power to be read off at a glance when the revolutions and angle of torque are known. The revolutions are usually readily obtainable by counter or tachometer, and it only remains to ascertain the torque, which must be done with extreme accuracy if the resulting horsepower is to be anywhere near the truth.

Torsion-Meters.—There are two well-known methods in use for measuring the torque of a revolving shaft: one electrical, the other mechanical. An example of the former method is the Denny-Johnson torsion meter. The mechanical method is represented by the Föttinger apparatus. It is not proposed to refer to these methods any further than to say that both involve certain intermediate links between the shaft under observation and the recording instrument, and that such links are liable, under varying conditions, to introduce errors.

The apparatus now to be considered abolishes these intermediate links, and measures the torsion directly from the shaft itself. As a beam of light is the essential feature of the apparatus, it has been called the "Flashlight" torsion meter, and will be so referred to in the following description. The intrinsic parts of the flashlight torsion meter consist of two thin discs secured to and revolving with the shaft, a fixed lamp, and a movable "torque finder." Figs. 8 to 13 show how these are disposed with relation to the shaft, and Figs. 14 and 15 are photographs of the apparatus as actually fitted.

The discs are similar, and each is perforated near its periphery by a small radial slot. The lamp has a mask perforated with a similar slot, and the torque finder has an eye-piece which can be moved circumferentially, and is also provided with a slot similar to the others. When the four slots are in the same radial plane, an observer looking through the eye-piece can see the beam from the lamp, and at every revolution of the shaft a flash is seen. The faster the shaft re-

volves the steadier the light appears, owing to the physical effect of persistence on the retina. The apparatus forms, in fact, a pair of photographic shutters, one shutter being at the lamp end and the other at the torque-finder end.

Immediately the shaft begins to transmit power the discs twist relatively to one another, and their slots are not now contained in the same radial plane. The beam still flashes through the shutter at the lamp end, but is obstructed at the torque finder, because of the relative displacement of the discs. To pick up the flash again it is necessary to move the eye-piece circumferentially by an amount equal to this displacement. Attached to the eye piece is a vernier, which moves over a fixed graduated scale on the torque finder, and enables the angle of torsion to be read off in degrees.

The width of the slots is eliminated by utilizing their edges, Fig. 8, and noting carefully the exact instant that the last ray is cut off as the eye piece is moved over. This gives an infinitesimally fine radial line of light as datum, and enables the torque to be measured to $\frac{1}{100}$ deg., or $\frac{1}{36000}$ of the circumference. No difficulty is experienced in attaining this degree of accuracy. Several independent observers have on occasion obtained exactly the same reading on setting the torque finder for themselves, an operation which takes only a few seconds to perform. To illustrate the appearance of the flash, as viewed through the torque finder, and also the method of measuring the torque and ascertaining the power, an image will now be projected on the lantern screen.

We will suppose, first, that the shaft is revolving idly; the propeller is disconnected; or, if in a turbine vessel, the steam supply may be shut off for a few moments and the turbine opened to the condenser vacuum. The "way" of the vessel will then cause the propeller to revolve the shafting and turbine without appreciable torque. By moving the torque finder over until the flash is just cut off, we obtain the "zero" reading, which must, of course, be subtracted from any subsequent power readings. The zero in this case is, say, 0.56 degree. Now we will suppose that the shaft is transmitting

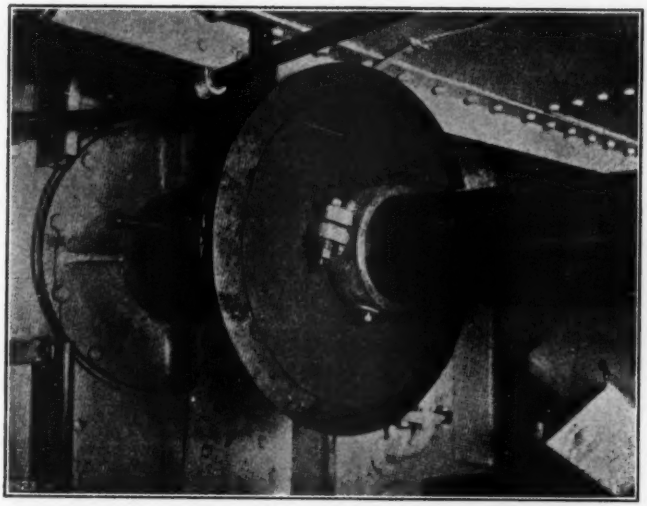


Fig. 14.

Fig. 15.

Fig. 16.

power, and has twisted the discs relatively to one another, thus obscuring the light. To pick up the flash again, the torque-finder eye piece, with its vernier, must follow the slot in its corresponding disc, and to get the exact angle of torque the point of cut-off is located as before. The vernier scale now shows 2.45 degrees, which, when the zero reading is subtracted, gives an effective torque of 1.89 degrees. To obtain the horsepower we refer to a power diagram similar to that shown in Fig. 7. Assuming that we know the revolutions to be 500 per minute, then in this case the horsepower is seen to be 3,820.

In Appendix II are set out in progressive form some shaft horsepowers recently obtained on a three-shaft turbine steamer by means of the flashlight torsion meter. The results extend over a series of trials carried out under varying conditions, the total powers recorded ranging from 7,975 down to 37. It will be seen that the port low-pressure turbine indicates less power than the starboard throughout the series. On investigation it was found that the blade-tip clearances differed slightly, the percentage difference corresponding almost exactly with the percentage difference of the powers.

In a turbine installation the turning moment is so uniform that it is perhaps only necessary to indicate the torque at one point of the revolution. But in shafting driven by reciprocating engines the turning moment is anything but uniform, and a torque reading taken at only one point in the revolution might give an entirely misleading result. The torque should be ascertained at several points in the circle, and plotted out on a development of the circumference to obtain the mean turning moment.

To meet this condition the flashlight torsion meter is slightly modified, as shown in Fig. 13. Instead of having only one slot at a fixed radius, each disc is perforated with several slots, corresponding to the number of points in the revolution at which it is desired to ascertain the torque, the slots being disposed in the form of a spiral at varying radii. The outermost slot, for instance, may indicate the torque when

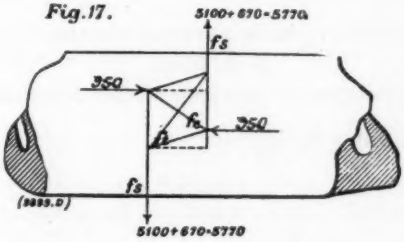
the high-pressure crank is on its top center, and so on. The light and the eye piece of the torque finder must be moved radially to or from the shaft to come opposite each pair of slots in the discs, and observations made, as before described, for each position.

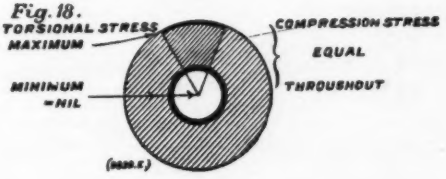
A torsion-meter record taken from a reciprocating-engine shaft by this method, and a set of indicator diagrams obtained simultaneously from the engine, would provide an interesting means of comparison, and very shortly such a comparison may be available. Of perhaps greater interest would be a comparison between the torsion-meter records of two identical vessels run at the same speed, one of which is propelled by turbines, the other by reciprocating engines; as, for example, in the Cunard liners, *Carmania* and *Caronia*, or H. M. S. *Amethyst* and *Topaze*. We should then have the exact power transmitted to the propellers of each vessel respectively, the only disturbing element in an exact comparison being the vexed question of propeller efficiency. But a consideration of the compound stresses in propeller shafting suggests that this problem might be solved by the use of some simple apparatus that would record the actual thrust along the line of shafting, and enable a direct comparison to be instituted between the power transmitted to, and the thrust produced by, the propeller. This, however, would naturally form the subject of another paper.

NOTE.—Besides the figures and diagrams appended, several lantern slides, some being mechanical, were projected on the screen, and added considerably in conveying to the minds of the audience the principles and methods of working the flash-light torsion meter. A working model of the apparatus was on exhibition, together with actual specimens of discs, lamps, torque finders, &c., all of which evoked the interest of members present.

But the above applies only to the outer skin of the shaft when the twisting stress is a maximum (Fig. 18). The twisting stress at the center is *nil*, whilst the compression stress is uniformly distributed over the whole area of section. There-

fore the effect of compression in weakening the shaft's resistance to torsion will be less than the percentage above quoted.





If the effect at the surface is 11½ per cent., and at the core *nil*, we may assume a mean effect over the whole section to be 0.5 of 11½ per cent. = 5¾ per cent. for a solid shaft; and 0.75 of 11½ per cent. for the hollow shaft in question = 8.6 per cent. for a hollow shaft of normal proportions. The true modulus of rigidity for calculating power would then be 8.6 per cent. less than modulus for torsion alone. Actual experiments, however, show only about half of this effect (see diagram, Fig. 5).

APPENDIX I.

Investigation for Torsion Test of Shafting as Described in Figs. 2 to 5.

Shaft, $7\frac{1}{2}$-in. in external diameter.
$3\frac{5}{8}$-in. hole.
Length between pointers, 20 feet 6 inches.

Maximum deflection in degrees

$$= \frac{H.P. \times C \times L''}{(D^4 - d^4) \times \text{revs.}} = \frac{5{,}000 \times 0.95 \times 3.27 \times 246}{2{,}997 \times 750} = 1.7 \text{ degree.}$$

($C = 3.27$ for a modulus of 11,250,000),

and maximum thrust

$$= \frac{0.65 \times 5{,}000\ H.P. \times 33{,}000}{33 \text{ kts.} \times 101.3 \times 2{,}240} = 14.3 \text{ tons.}$$

Area of shaft = 33.7 sq. inches: $\frac{14.3 \times 2{,}240}{33.7} = 950$ pounds per square inch compressive stress.

Torsional (or shear) stress at full power—

$$T.\ M. = \frac{33{,}000 \times 0.95 \times 5{,}000\ H.P.}{750 \text{ revs.} \times 2\pi} = 3{,}300 \text{ lbs. at } 1' \text{ radius.}$$

$$T.\ M. = 0.196 \left(\frac{D^4 - d^4}{D}\right) f, \text{ therefore } f = \frac{33{,}300 \times 12 \times 7.5}{0.196 \times 2{,}997}$$

$= 5{,}100$ pounds per square inch torsional stress.

A shear stress produces equal tension and compression stresses at 45 degrees–*i. e.*, $f_s = f_t = f_c$. (Lineham.)

But the compressive stress due to propeller thrust augments the

$$f_c \text{ by } \frac{950}{\sqrt{2}} = 670 \text{ pounds per square inch,}$$

and as $f_s = f_c$, therefore 670 must be added to the 5,100+670 = 5,770 = total f_s = gross torsional (or shear) stress, and 670 = $11\frac{1}{2}$ per cent. of 5,770 (Fig. 17).

APPENDIX II.

Flashlight Torsion Meter.—Turbine Steamer.

Actual readings and corresponding horsepowers taken during trial trips under varying conditions of displacement and propellers.

Turbine shaft.	Degrees torque.	Revolutions per minute.	Shaft horsepower.	Shaft horsepower. Total.
Starboard low-pressure	1.43	482.9	2,775	7,975
Center high-pressure	1.69	461.2	2,600	
Port low-pressure	1.37	472.8	2,600	
Starboard low-pressure	1.31	461.2	2,410	6,940
Center high-pressure	1.65	426.8	2,330	
Port low-pressure	1.24	457.3	2,200	
Starboard low-pressure	1.15	426.4	1,970	5,960
Center high-pressure	1.51	417.6	2,080	
Port low-pressure	1.13	418.9	1,910	
Starboard low-pressure	1.05	418.4	1,765	5,555
Center high-pressure	1.52	422.3	2,120	
Port low-pressure	1.02	415.5	1,670	
Starboard low-pressure	0.21	198.6	162	495
Center high-pressure	0.27	206.3	185	
Port low-pressure	0.19	183.5	148	
Starboard low-pressure	0.22	146.7	88	257
Center high-pressure	0.21	171.4	87	
Port low-pressure	0.13	144.8	82	
Starboard low-pressure	0.07	46.3	13	37.2
Center high-pressure	0.05	86.1	15	
Port low-pressure	0.01	24.4	9.2	

U. S. S. *KANSAS.*

DESCRIPTION AND OFFICIAL TRIALS.

By WILLIAM ASHLEY LEAVITT, JR., ASSOCIATE.

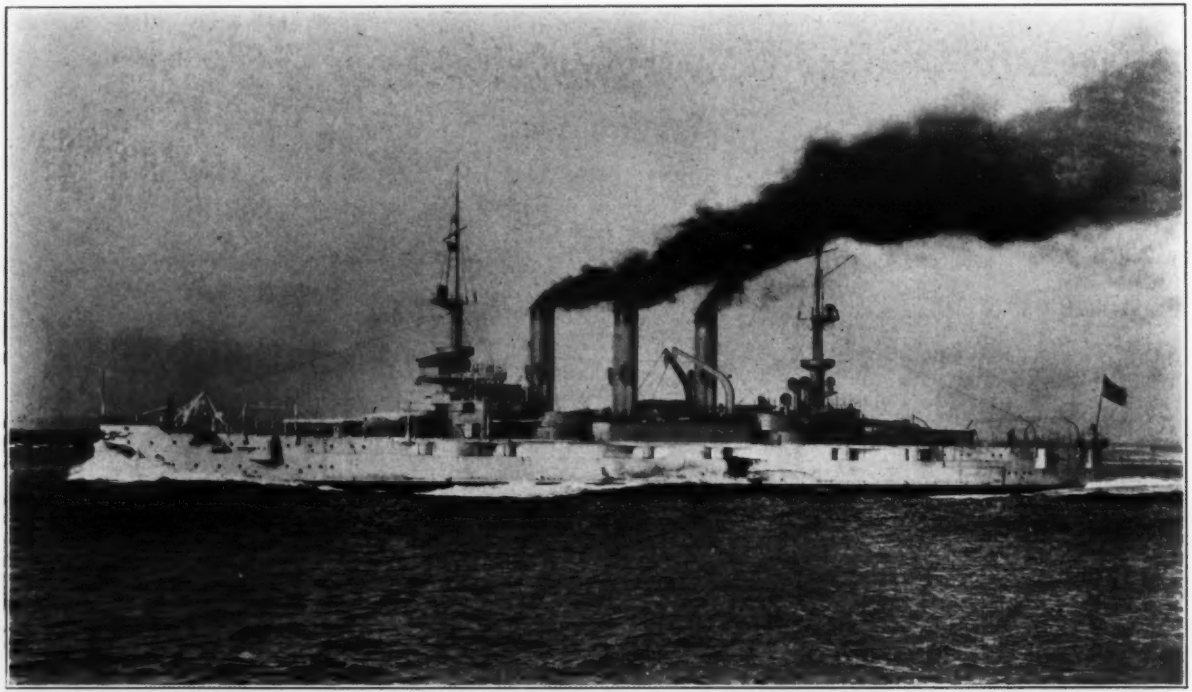
The *Kansas* (Battleship No. 21) building by The New York Shipbuilding Company, Camden, New Jersey, is one of three ships of this class authorized by Act of Congress, March 3d, 1903, the sister ships being the *Minnesota* and *Vermont.*

The contract was signed July 16th, 1903; keel laid February 10th, 1904; hull launched August 12, 1905; and the vessel was delivered to the Government April 15, or four months over the contract time, forty-two (42) months; the contractors claiming non-receipt of armor and certain details of ordnance as the principal causes of delay.

The contract price for the construction of hull and machinery was $4,165,000, of which sum $3,165,000 was allotted to hull and $1,000,000 for the propelling machinery and auxiliaries. The contract required a speed of 18 knots to be maintained for four consecutive hours on a mean draught of 24 feet 6 inches, corresponding to a displacement of 16,000 tons, and an additional endurance run for twenty-four consecutive hours at not less than 13,200 I.H.P., equivalent to eighty per cent. of the designed H.P.

The contract allotment of weights for machinery was 1,500 tons; the actual weight being about 1,504 tons, or about 4 tons overweight.

The propelling machinery was given dock trials as follows: Starboard engine, November 12th and 13th, 1906; aggregate number of hours run, 16; maximum revolutions, 77. Port engine, November 14th, 15th, 16th, 1906; aggregate number of hours run, 24; maximum revolutions, 78; the results being



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U. S. S. "Kansas."

generally satisfactory, the only alterations of note being the addition of heavy cast-steel brackets resting on the inner bottom to support the overhang of the long main bearings between the I.P. and after L.P. cranks, and an increase in the size of the relief valves on the main feed discharge.

OFFICIAL TRIALS.

The Navy Department having given permission to begin the official trials, December 13th, 1906, the ship left Camden at 1 P. M., December 5th, proceeding to Boston for docking at the Navy Yard, reaching that place at 8 A. M., December 9th, or 91 hours out from Camden. Fog in the Bays at both ends of the trip, and a strong head wind with a fairly heavy sea running being the principal causes of delay. The ship was in dock exactly fifty hours, leaving for Rockland at 6 A. M., December 12th, and anchoring off Owls Head Light at the entrance to Rockland Harbor at 4·30 P. M. This run being made under the first favorable weather conditions experienced since leaving Camden at 2 P. M., a run at maximum power was begun, but after running 55 minutes it was called off on account of the after main bearing of the port engine showing signs of heating. This was the only speed run attempted by the contractors before the Trial Board took charge and the official trials were made. December 13th being favorable both as regards wind and sea, the standardization runs were made over the Government measured-mile course off Monhegan Island, the maximum speed attained being at the rate of 18.499 knots at 122.55 revolutions, developing 20,373 I.H.P.; and the average of the five runs at full power was 18.094 knots at 121.43 revolutions, developing 19,410 I.H.P.

On December 14th the four-hour endurance run in open sea was made successfully, the course being laid along the Maine coast, and headed toward Boston Light. The most remarkably close engine running was maintained throughout the four hours, the average variation in the revolutions being less than one-half of a revolution. The average number of

revolutions for the starboard engine was 122.12; port engine, 130.52, and average of both engines, 121.32, developing 1,930 I.H.P. for the main engines, or a total for all machinery of 19,757 I.H.P. The average speed was 18.094 knots, being exactly the same as the average of the five full-power standardization runs; average slip of the propellers being 14.86.

The performance of the machinery, both main and auxiliary, was generally satisfactory, except in details as follows: The port after main bearing had to be watched very closely, as it gave frequent evidences of heating, and an excessive amount of oil had to be used on all bearings, due, undoubtedly, to the excessive load and strains on the working parts on account of the engines being called upon to develop about 20 per cent. above the designed power.

The port H.P. and forward L.P. crank pins heated during the last 15 minutes, requiring the use of water, and the H.P. and I.P. crank pins and crossheads of both engines pounded considerably. When called upon to deliver considerable feed the main feed pumps pounded a good deal, which was undoubtedly due to the suction pipes being too small.

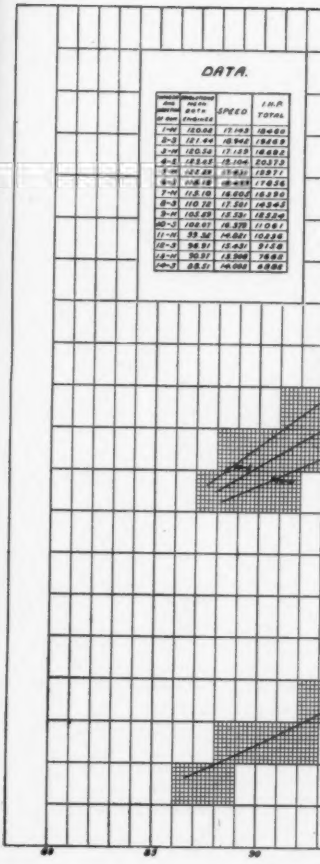
STANDARDIZATION.

For the purpose of standardizing the screws, runs were made over the measured-mile course off Rockland, Maine, Dec. 13, 1906. The weather was clear and cool with moderate to gentle breezes. Smooth sea.

Previous to the vessel's starting on the standardization runs the draught was taken and found to be—

Forward, feet and inches, ...	24–08½
Aft, feet and inches, ...	24–05⅝
Mean, feet and inches, ...	24–07 1/16
Corresponding displacement, tons, . . .	16,085

These runs gave the data from which the curves on Plate I were plotted.


DATA.
SPEED
I.H.P. TOTAL
1-M 120.04 17.143 18460
2-3 121.44 16.962 19469
4-M 120.52 17.159 18682
6-5 123.65 15.104 20373
8-M 122.89 17.631 19971
9-3 118.18 16.459 17836
7-M 115.10 16.206 18390
6-3 110.72 17.261 19342
5-M 105.29 15.324 18224
10-3 104.01 16.378 11061
11-M 99.36 14.821 10246
12-3 96.91 15.451 9128
13-M 90.37 15.908 7698
14-3 88.51 14.050 6868
88
83
30

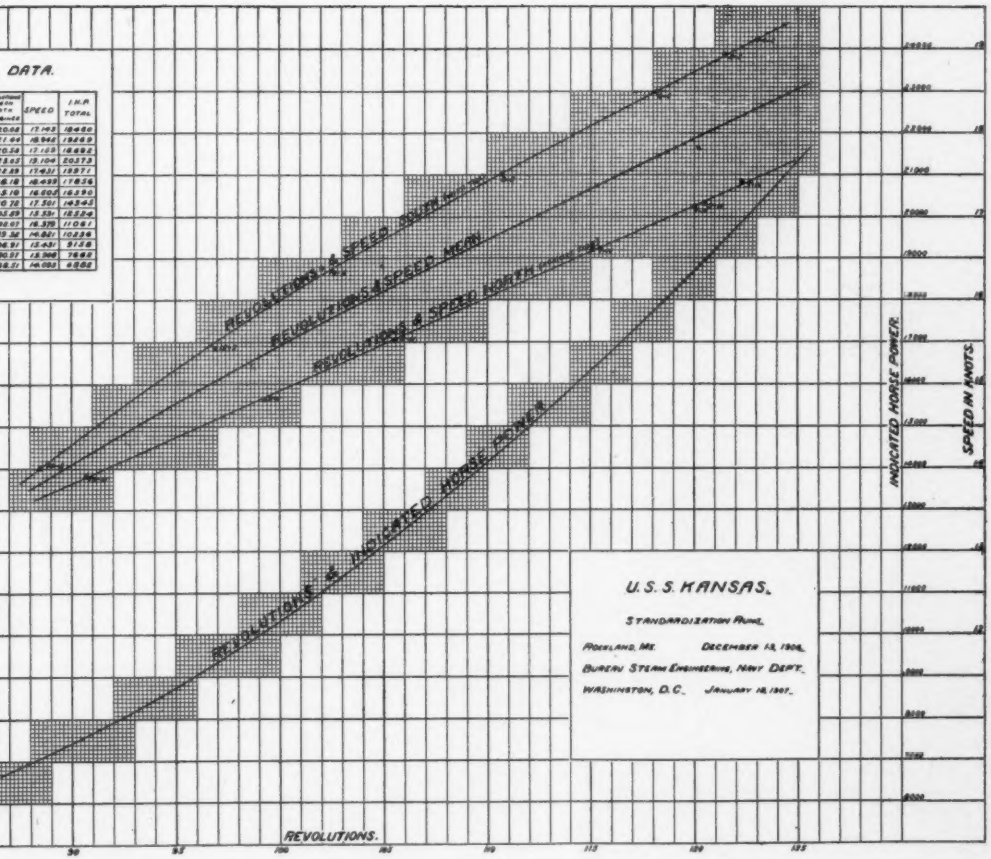

DATA.
SPEED
I.H.P. TOTAL
REVOLUTIONS & SPEED OVER
REVOLUTIONS & SPEED NORTH
REVOLUTIONS & INDICATED HORSE POWER
U.S.S. KANSAS.
STANDARDIZATION RUNS.
PORTLAND, ME. DECEMBER 13, 1906.
BUREAU STEAM ENGINEERING, NAVY DEPT.
WASHINGTON, D.C. JANUARY 16, 1907.
INDICATED HORSE POWER.
SPEED IN KNOTS.
REVOLUTIONS.

Plate I.

After the standardization run the draught was—

Forward, feet and inches, ...	24–07⅜
Aft, feet and inches, ...	24–05⅞
Mean, feet and inches, ...	24–06⅛
Corresponding displacement, tons, . . .	16,064

FOUR-HOURS' OFFICIAL TRIAL.

The ship left her anchorage at Rockland, Maine, at 8·00 A. M. December 14, 1906, and steered a straight southeast course out to sea. A period of two hours was consumed in working the machinery up to power, and at 9·51 the signal was given that the four-hour run was begun, which was finished at 1·51 P. M. without the least trouble arising to cause a pause for an instant. The main engines ran with remarkably little vibration, the smooth running of the valve gear, which seemed to almost float, being particularly noticeable. The water service being used only on the crosshead guides and thrust bearings, furnishes satisfactory evidence of good running and workmanship.

At the conclusion of the four-hour run the engines were slowed down for maneuvering the ship. The time required to put the helm from hard-a-starboard to hard-a-port was ten seconds, and from hard-a-port to hard-a-starboard was ten seconds. During this test the maximum angle of heel was seven degrees.

In the firerooms during the four-hour run everything worked easily and with great regularity. The time-firing device installed by the contractors, the design of two men in their employ, regulated the intervals of firing, which averaged five minutes for each furnace. The performance of this device was most satisfactory and the simplicity of design makes it easy to regulate, overhaul and stow in the ship. The forced-draft blowers and engines ran with noted smoothness, and at no time did the revolutions exceed 340 per minute. In the foreward firerooms at intervals the auxiliary feed pumps were used to make up feed, the main pumps not being able to

reach those boilers when forced. In order to determine accurately the amount of coal consumed it was put into bags holding 200 pounds each, and a record kept of the number of bags used.

The number of men used in each fireroom was as follows:

1 Water tender.
4 Firemen.
2 Coal passers in fireroom.
2 Coal passers in bunkers.
1 Apprentice boy.

DATA FOR FOUR-HOUR RUN.

Draught at beginning of trial, forward, feet and inches	24–07⅛
aft, feet and inches	24–07⅜
Mean draught at beginning of trial, feet and inches	24–07¼
Corresponding displacement at mean draught at beginning of trial, tons	16,103
Mean draught at end of trial, not taken, feet and inches, (5·00 P. M.)	24–05½
Corresponding displacement at mean draught during trial, tons	16,000
Speed of ship, in knots per hour	18.094

	Starboard.	*Port.*
Slip of propellers, in per cent. of their speed	15.43	14.29

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	278.2	
engine (per gauge), pounds	264.5	
H.P. steam chest (gauge), pounds	250.0	250.0
1st receiver (absolute), pounds	105.0	108.0
2d receiver (absolute), pounds	38.0	38.0
Vacuum in condensers, inches of mercury	28.0	27.9

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	44.0	44.4
Discharge, degrees	101.0	101.6
Hotwell, degrees	69.0	69.2
Feed water, degrees	165.0	164.4
Engine room, upper platform, degrees	86.0	89.5
working platform, degrees	74.0	74.0
Firerooms, above grating, degrees	68.0	
working level, degrees	65.7	
Atmosphere	32.5	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	*Port*
Main engines (mean for trial)	122.12	120.52
average for both	121.32	
Pumps, main, air	19.0	18.0
circulating	140.0	133.0
feed	27.0	32.0
bilge	64.0	43.0
auxiliary feed	.....	.....
condenser	13.0	36.0
Blower engines	314.0	
Speed of ship, in knots per hour	18.004	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	15.43	14.20
Air pressure in firerooms, inches of water	.596	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	123.5	122.5
I.P. cylinder	49.4	49.8
F.L.P. cylinder	20.7	20.2
A.L.P. cylinder	20.9	21.0
Mean equivalent pressure, in pounds per square inch referred to combined area of L.P. pistons	56.61	56.46

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	2,958.0	2,896.0
I.P. cylinder	3,194.0	3,183.0
F.L.P. cylinder	1,778.0	1,711.0
A.L.P. cylinder	1,797.0	1,785.0
Total	9,727.0	9,575.0
both engines	19,302.0	

AUXILIARIES.

Air pumps	13.7	13.3
Circulating pumps, main	52.3	44.5
Feed pumps, main	55.6	64.1
Bilge pumps	9.8	3.9
Water-service pumps	...	...
Auxiliary condenser pumps	1.4	4.1
Blower engines (forced draft)	97.6	
Steering engines (estimated)	5.0	
Dynamo engines	90.0	
Total	455.0	
Total all machinery in operation	19,757.0	

COAL.

Kind and quality	Pocahontas, hand-picked, excellent.
Average thickness of fires, inches	10–12
Coal burned per hour, pounds	34,328.0
Average temperature of atmosphere, degrees Fahrenheit	32.5
air pressure in firerooms, inch of water	.596

Deduced Data.

Coal burned per square foot of G.S. per hour, pounds	31.21
H.S. per hour, pound	.651
I.H.P. of main engines per hour, pounds	1.779
total I.H.P. main engines and all auxiliary engines per hour, pounds	1.737

INDICATOR CARDS.

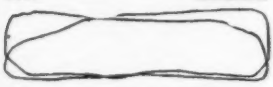
One set of cards representing maximum conditions obtainable under forced draft without steam to receivers were selected for comparison with theoretical cards and average condition on the trial. See the accompanying sketch of the cards and the same combined, Plate II.

Port Engine.	Cards taken witho't steam to receivers. M.E.P.	Cards taken on trial with steam to receivers. M.E.P.
H.P. cylinder, average	125.0	.....
I.P. cylinder, average	50.15	.....
F.L.P. cylinder, average	20.35	.....
A.L.P. cylinder, average	21.39	.....
Starboard Engine.		
H.P. cylinder, average	125.3	.....
I.P. cylinder, average	48.65	.....
F.L.P. cylinder, average	20.79	.....
A.L.P. cylinder, average	21.01	.....

	H.P.	I.P.	F.L.P., A.L.P. Combined.
Area of piston, top, square inches	829.57	2,206.19	5,844.94
Ratio of areas	1.0	2.66	7.05
Per cent. cut off, top, act. average	81.4	78.25	75.75
bottom, act. average	74.25	70.55	64.42
clearance, top, act. average	23.82	17.43	15.32
bottom, act. average	21.81	19.42	17.47

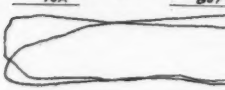
PORT MAIN ENGINE.

H.P. CYLINDER.
I.H.P. = 2069
H.P. REC. GAGE = 260 — SCALE SPRING = 120
M.E.P. = 124.6 — M.E.P. = 124.9
TOP — BOTTOM

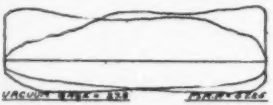


I.P. CYLINDER.
I.H.P. = 3220
I.P. REC GAGE = 98.8 — SCALE SPRING = 80
M.E.P. = 49.85 — M.E.P. = 50.44
TOP — BOTTOM

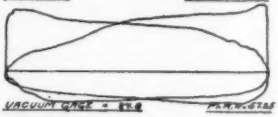
M.P. CYLINDER.
I.H.P. = 3009
M.P. REC. GAGE = 260 — SCALE SP
M.E.P. = 124.4 — M.E.P. =
TOP — BOT



FOR'D L.P. CYLINDER.
I.H.P. = 1730
F.L.P. REC. GAGE = 88.8 — SCALE SPRING = 20
M.E.P. = 20.61 — M.E.P. = 20.2
TOP — BOTTOM



AFT. L.P. CYLINDER.
I.H.P. = 1846
A.L.P. REC. GAGE = 93.8 — SCALE SPRING = 20
M.E.P. = 21.87 — M.E.P. = 21.51
TOP — BOTTOM



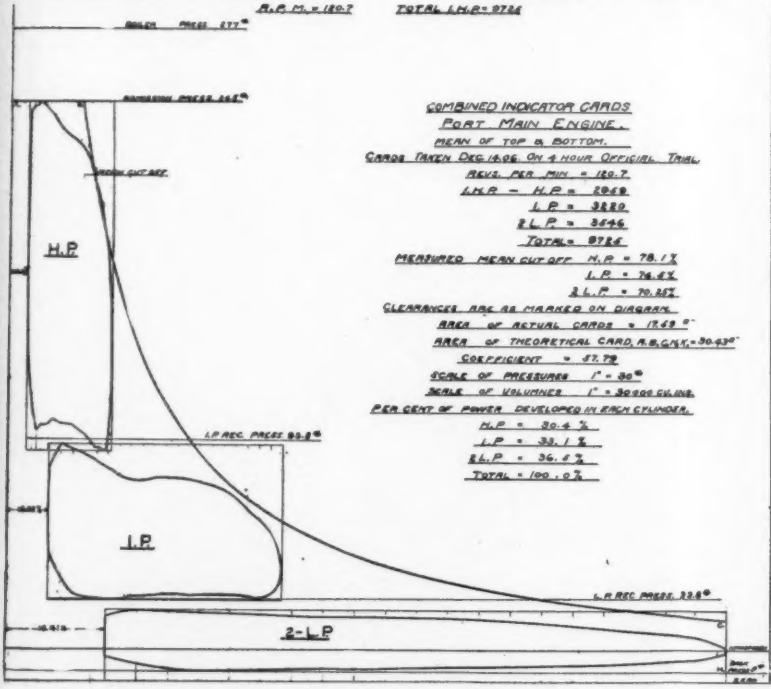
FOR'D L.P. CYLINDER.
I.H.P. = 1787
F.L.P. REC. GAGE = 93 — SCALE SP
M.E.P. = 20.71 — M.E.P.
TOP — BOT

VACUUM GAGE = 13.8 — MEAN VACUUM 13.8

VACUUM GAGE = 13.8 — MEAN VACUUM 13.8

VACUUM GAGE = 13.8 — MEAN VACUUM

I.H.P. = 9724



COMBINED INDICATOR CARDS
PORT MAIN ENGINE.

MEAN OF TOP & BOTTOM.

CARDS TAKEN DEC. 18.06. ON 4 HOUR OFFICIAL TRIAL.

REVS. PER MIN. = 120.7

I.H.P. — H.P. = 2069
I.P. = 3220
2 L.P. = 3646
TOTAL = 9724

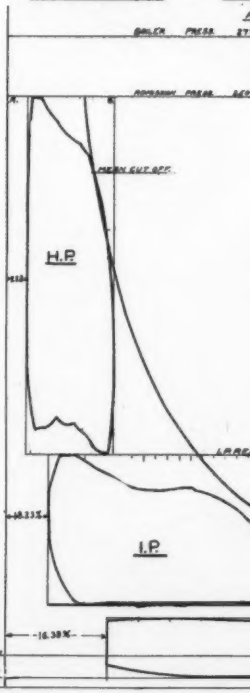
MEASURED MEAN CUT OFF H.P. = 78.1%
I.P. = 74.4%
2 L.P. = 70.25%

CLEARANCES ARE AS MARKED ON DIAGRAM.

AREA OF ACTUAL CARDS = 17.69 □"
AREA OF THEORETICAL CARD, A.B.C.N.K. = 30.93 □"
COEFFICIENT = .57.72
SCALE OF PRESSURES 1" = 80#
SCALE OF VOLUMES 1" = 30.000 CU. INS.

PER CENT OF POWER DEVELOPED IN EACH CYLINDER.

H.P. = 20.4 %
I.P. = 33.1 %
2 L.P. = 36.5 %
TOTAL = 100.0 %

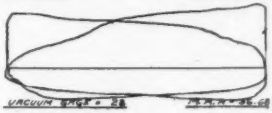


STARBOARD MAIN ENGINE.

H.P. CYLINDER.
I.H.P = 3008
SCALE SPRING: 140
M.E.P = 126.2
BOTTOM.

I. P. CYLINDER.
I.H.P = 3168
I.P REC. GAGE = 26
SCALE SPRING: 80
M.E.P = 48.96
TOP
M.E.P = 48.96
BOTTOM.

AFT. L.P. CYLINDER.
I.H.P = 1808
A.L.P REC GAGE = 22
SCALE SPRING: 20
M.E.P = 20.91
TOP
M.E.P = 21.22
BOTTOM.



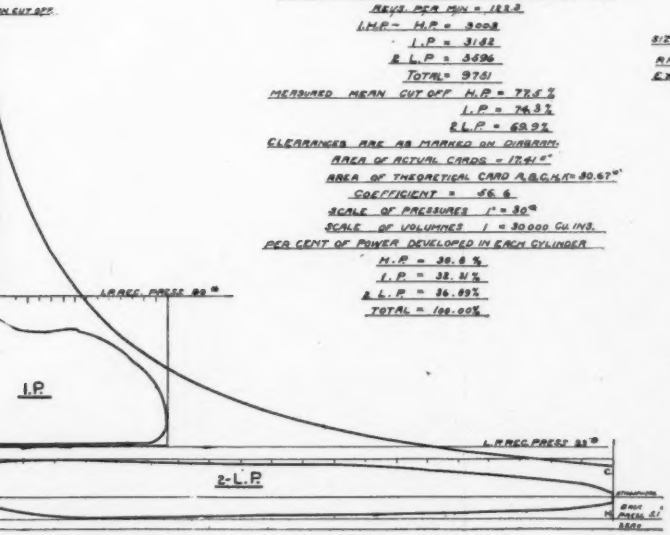
VACUUM GAGE = 28

NOTE:
CARDS WERE TAKEN BY THE TRIAL BOARD ON THE 4 HOUR OFFICIAL TRIAL DEC 14-06. THROTTLE FULL OPEN, LINKS FULL GEAR, BY-PASS CLOSED.

R.P.M. = 122.3
TOTAL I.H.P = 9761

COMBINED INDICATOR CARDS.
STARBOARD MAIN ENGINE.
MEAN OF TOP & BOTTOM.
CARDS TAKEN DEC 14-06 ON 4 HOUR OFFICIAL TRIAL
REVS. PER MIN = 122.3
I.H.P = H.P = 3008
I.P = 3168
2 L.P = 3696
TOTAL = 9761
MEASURED MEAN CUT OFF H.P = 77.6 %
I.P = 74.3 %
2 L.P = 69.9 %
CLEARANCES ARE AS MARKED ON DIAGRAM.
AREA OF ACTUAL CARDS = 17.41 "
AREA OF THEORETICAL CARD A,B,C,H,K = 30.67 "
COEFFICIENT = .56.6
SCALE OF PRESSURES 1" = 30#
SCALE OF VOLUMES 1 = 30.000 CU.INS.
PER CENT OF POWER DEVELOPED IN EACH CYLINDER
H.P = 30.8 %
I.P = 32.4 %
2 L.P = 36.89%
TOTAL = 100.00%

SIZE OF ENG.^{s} $\frac{32\frac{1}{2}" \times 43" \times 2-61"}{48"}$
RATIO OF CYLINDERS 1 : 2.66 : 7.04
EXPANSIONS 9.04



NSAS."——**Plate II.**

Volume at cut off, top, H.P. cylinder, cubic inches	32,420.0
of clearance, top, H.P. cylinder, cubic inches	9,471.68
at cut off, total H.P. cylinder, cubic inches	41,891.68
bottom H.P. cylinder, cubic inches	28,150.0
of clearance, bottom, H.P. cylinder, cubic inches	8,256.11
at cut off, total, H.P. cylinder, cubic inches	36,406.11
Mean of top and bottom volumes H.P. cylinder, cubic inches	39,148.89
Piston displacement and clearance, top, L.P. cylinders, cubic inches	323,531.78
Piston displacement and clearance, bottom, L.P. cylinders, cubic inches	224,898.17
Piston displacement and clearance, mean, L.P. cylinders, cubic inches	324,214.97
Ratio of initial to final volumes of steam	8.28
Areas of actual trial cards	17.5
theoretical cards	30.55
Ratio of actual to theoretical	.5729

TWENTY-FOUR HOURS' OFFICIAL TRIAL.

On December 15th, at 7 A. M., the ship got under way for the twenty-four-hour endurance run at sea, and the run was begun at 8·20 o'clock, ending at that hour December 16th. The course was from Boston to Cape May, and conditions as regards wind and sea very favorable.

The average speed during the entire run was 16.65 knots, at 109.53 revolutions and 14,532 I.H.P., with a total of 14,532 I.H.P. for all machinery. The draught (estimated) was 24 feet, and the displacement 15,620 tons. Mean slip of propellers, 13.24. Except as noted, the performance of the machinery was very satisfactory. During the last hour the port I.P. crank-pin brasses heated, requiring the use of water, water also being used on the valve guides, I.P. crank pin, as a precautionary measure. Boiler "K" was cut out during the last two hours, on account of a leak in the front cross-connection box. The leak was due to faulty welding, and the box was replaced by the Babcock & Wilcox Co.

DATA OF TWENTY-FOUR HOURS' ENDURANCE TRIAL FROM BOSTON, MASS., TO DELAWARE BREAKWATER, DECEMBER 15, 1906.

	Starboard.	*Port.*
Maximum average revolutions per minute for 15-minute period	112.1	112.6
Average revolutions per minute for 24 hours	109.7	109.36
Mean revolutions per minute for both engines	109.53	

	Starboard.	*Port.*
Maximum steam pressure at boilers, pounds	273.0	
Average steam pressure at boilers, pounds	263.0	
Maximum steam pressure at engines, pounds	253.0	254.0
Average steam pressure at engines (H.P. steam chest), pounds	211.0	220.0
Maximum air pressure in firerooms, inch	.8	
Average air pressure in firerooms, inch	.52	
Maximum vacuum, each engine, inches	27.0	28.0
Average vacuum, each engine, inches	26.8	28.0
Collective I.H.P. of all main engines	7,063	7,091
main engine, air, circulating feed and hot-well pumps	14,332	
Collective I.H.P. of main and auxiliary engines during trial	14,532	
Kind and quality of coal used on trial*	Pocahontas, run of mine.	
Draught at beginning of trial (forward), feet and inches,	24-2½	
(aft), feet and inches	24-4	
Mean draught at beginning of trial, feet and inches	24-3¼	
Corresponding displacement at mean draught at beginning of trial, tons	15,810	
Draught at end of trial	Not taken, ship at sea.	
Speed of ship, knots per hour	16.65	
Slip of propellers, in per centum of their own speed	13.35	13.13

INSPECTION AFTER TRIAL.

SUMMARY OF THE FINDINGS OF THE BOARD.

Starboard Engine.—Examined Nos. 3, 5 and 6. Main bearings and all caps in good condition except No. 5, which was slightly scored and dragged, and No. 6 cap dragged and white metal slightly cracked at the crown. Rolled out bottom brass of No. 6 and found white metal slightly dragged and journal roughened at forward end.

Examined steam and water ends of main air pump; steam end good, but water end showed considerable grease.

Examined steam and water ends of main feed pumps and found them in good condition.

Examined H.P. and I.P. and forward L.P. crank pins and brasses and found them in good condition.

Examined feed tank and found it clean.

---

*Average quantity of coal used per I.H.P. per hour not obtainable, there being no observers.

Examined after L.P. crosshead brasses and pins and found them in good condition.

Examined main condensers, which were found to be tight; and fresh-water side, so far as could be seen, was clean, but a few tubes were partly filled with dirt.

Examined all main engine cylinders and H.P. ring and springs and fonnd them in good condition.

Examined H.P., I.P. and forward L.P. valves and found them in good condition.

Port Engine.—The following parts were examined and found in good condition: Main engine cylinders, H.P. valve chest and valve; I.P. valve chest and valves; A.L.P. valve chest and valves, except the H.P. cylinder's, which was cut in one place near the top, and apparently corresponding with a porous spot on the piston ring. The main condenser, feed and filter tank, main air pump, and main feed pumps were found in good condition.

The A.L.P. crosshead brasses and pin were examined and found in good condition. The H.P., I.P. and A.L.P. crank pins were examined and found as follows: The H.P. crank-pin brasses show signs of slight dragging of the white metal, and the white metal is slightly cracked in the crown of the crown brass. The I.P. pin heated on the twenty-four hour run; the white metal in the brasses shows signs of running and dragging, and the A.L.P. pin brasses do not fit properly.

All pins in good condition.

Crank-shaft bearings Nos. 3, 5 and 6, were examined and found in good condition, except No. 6 bearing, which was found slightly cut at the after end for about 1½ inches. This bearing gave considerable trouble previous to the trials, and was refitted twice.

The Board decided that the H.P. pistons should have new packing rings, made solid and without springs; that all crank pins, crank-shaft bearings and crossheads should be readjusted; that the I.P. crank-pin brasses should be relined with white metal; that No. 6 crank-shaft bearing should be refitted, and

that all crank pins and journals should be stoned in places where nicks and scores show.

The steam cylinders and bearings of one blower engine were examined and found in excellent condition; and, as no trouble of any kind was experienced, even at high speed during the trials, examination of the other engines was not considered necessary.

All furnaces and drums or the boilers were carefully inspected; and found in good condition.

Tubes (4-inch) above the furnaces were generally bowed, the maximum bend being about $\frac{5}{16}$-inch degree, bends being upward as well as downward, but a number of tubes were straight.

Evidence of oil, to a very limited extent, was found in all of the boilers.

Five 4-inch tubes which carry the horizontal baffles in the outboard furnace showed signs of bulging, and the Board recommended that they be cut out.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet and inches	450- 0
on load water line (24 feet, 6 inches), feet and inches	450- 0
over all, feet and inches	456-04
of straight keel, feet and inches	362- 0
Projection forward of F. P., feet and inches	6-04
aft of A. P., feet and inches	0.00
Breadth, extreme, at L. W. L., outside of armor, feet and inches	76-10
molded, feet and inches	76-05$\frac{1}{2}$
Depth, molded, main deck, at side, M. S., feet and inches	43-01$\frac{3}{8}$
upper deck, at side, M. S., feet and inches	50-07$\frac{3}{8}$
Ratio of length to beam	5.94
Draught, L. W. L., for calculations, feet and inches	24-06
mean, on trial, feet and inches	24-05$\frac{1}{2}$
Displacement, normal draught, tons of S. W.	16,003
mean, on trial, tons of S. W.	16,000
per inch, at L. W. L., tons of S. W.	63.2
Area of midship section, square feet	1,811
L. W. L. plane, square feet	26,590
wetted surface, square feet	44,000

Center of gravity of L. W. L. plane, aft of M. S., feet	3.39
buoyancy above base line, feet and inches	13-04
aft of M. S., feet and inches	2-08
Transverse metacenter above center of buoyancy, feet and inches	18-09
Longitudinal metacenter above center of buoyancy, feet and in	532-0
Coefficient of fineness, block	.652
M. S	.962
L. W. L.	.769
cylindrical	.687
Number of frames	112
watertight compartments	324

Heights above L. W. L. (which is 24 feet 4⅝ inches above the molded base, the molded base line being 1⅛ inches above bottom of keel).

Top of truck light, foremast and mainmast, feet and inches	131- 0
Floor of range-finder platform, foremast and mainmast, feet, and inches	54-11
Searchlight platform, foremast, feet and inches	76- 1
mainmast, lower, feet and inches	43- 2½
upper, feet and inches	76- 1
Conning tower top at center line, feet and inches	42- 2
Flying bridge at side (top of plank), feet and inches	44- 1½
Forward bridge at side, top of beams, feet and inches	34- 1½
After bridge at side, top of beams, feet and inches	33-10½
Top of main-deck planking at stern, feet and inches	20-10½
breakwater at stern, feet and inches	23- 6
beading at stern, feet and inches	22- 1
main-deck planking at stern, feet and inches	20- 4½
flat armor deck, feet and inches	3- 0

MAIN BATTERY.

No. of Guns.	Caliber.	Location.		Train.		Gun No.		Remarks.
		Deck.	F'me.			S.	P.	
				Degrees.	*Degrees.*			
2	12-inch B.L.R.	Main	29	135 S.	135 P.	1	2	In turret.
2	12-inch B.L.R.	Main	85½	135 S.	135 P.	4	3	In turret.
4	8-inch B.L.R.	Gun...	44½	90 F.	55 A.	1-2	3-4	In turret.
4	8-inch B.L.R.	Gun...	69	90 A.	55 F.	5-6	7-8	In turret.
2	7-inch R.F.G.	Gun...	31	90 F.	26 A.	1	2	In sponson.
2	7-inch R.F.G.	Gun...	39	73½ F.	42½ A.	3	4	In sponson.
2	7-inch R.F.G.	Gun...	53	70 F.	46 A.	5	6	In sponson.
2	7-inch R.F.G.	Gun...	61	46 F.	70 A.	7	8	In sponson.
2	7-inch R.F.G.	Gun..	75	42½ F.	73½ A.	9	10	In sponson.
2	7-inch R.F.G.	Gun...	83	26 F.	90 A.	11	12	In sponson.

SECONDARY BATTERY.

No. of Guns.	Caliber.	Location. Deck.	Location. F'me.	Train.		Gun No. S.	Gun No. P.	Remarks.
				Degrees.	*Degrees.*			
2	3-inch R.F.G.	Gun...	16½	90 F.	30 A.	1	2	In sponson.
2	3-inch R.F.G.	Gun...	101	30 F.	90 A.	3	4	In sponson.
2	3-inch R.F.G.	Gun...	109	30 F.	90 A.	5	6	In sponson.
2	3-inch R.F.G.	Main..	52½	60 F.	60 A.	7	8	In casem'e.
2	3-inch R.F.G.	Main..	56½	60 F.	60 A.	9	10	In casem'e.
2	3-inch R.F.G.	Main..	61	60 F.	60 A.	11	12	In casem'e.
2	3-inch R.F.G.	Up'r...	36½	50 F.	50 A.	13	14	...
2	3-inch R.F.G.	Up'r...	50	50 F.	50 A.	15	16	...
2	3-inch R.F.G.	Up'r...	77½	50 F.	50 A.	17	18	...
2	3-inch R.F.G.	F'd bdg.	37	50 F.	50 A.	19	20	...
2	3-pdr. R.F.G....	Up'r ..	39½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Up'r...	75½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	F'd bdg.	40½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	77½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	79½	50 S.	50 P.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	75°	50 F.	50 A.	...	...	...
2	30-caliber.....	F'd bdg.	...	...	...	...	...	...

COAL-BUNKER CAPACITIES.

Measured at 6 inches below bottom of deck beams, and at 43 cu. ft. per ton.

Bunkers.	Location. Frame.	Location. Side.	Tons.
Above protective deck.			
B-119	40½-46	S.	61.9
B-118	40½-46	P.	61.9
B-121	46 -52	S.	70.6
B-120	46 -52	P.	70.6
B-123	52 -58½	S.	77.9
B-122	52 -58½	P.	77.9
B-127	58½-65	S.	76.9
B-126	58½-65	P.	76.9
B-129	65 -72	S.	78.2
B-128	65 -72	P.	78.2
Total.....			731.0
Below protective deck.			
B- 7	35 -40½	S.	120.0
B- 8	35 -40½	P.	131.7
B- 9	35 -40½	S.	50.5
B-10	35 -40½	P.	50.5
B-11	40½-45	S.	50.4
B-12	40½-45	P.	50.4
B-13	45 -49½	S.	69.4
B-14	45 -49½	P.	69.4
B-15	49½-54	S.	75.5
B-16	49½-54	P.	75.5
B-17	54 -58½	S.	77.5
B-18	54 -58½	P.	77.5
B-19	58½-63	S.	77.5
B-20	58½-63	P.	77.5
B-21	63 -67½	S.	76.0
B-22	63 -67½	P.	76.0
C- 5	67½-72	S.	59.2
C- 6	67½-72	P.	73.0
C- 7	67½-72	S.	49.3
C- 8	67½-72	P.	49.3
C- 9	72 -77	S.	51.5
C-10	72 -77	P.	49.5
C-11	77 -83	S.	46.2
C-12	77 -83	P.	46.2
Total.....			1,629.5

Recapitulation, Bunker Capacity.

Above protective deck.....	731.0
Below protective deck.....	1,629.5
Total.....	2,360.5

TANKS.

Capacities of Fresh-Water Tanks.

Tank No.	Compart. number.	Between frames.	Location.	Side.	Capacity.	
					Gallons.	*Tons.*
9	D-11	98-101	Hold, aft.....	S.	2,230	8.3
10	D-11	98-101	Hold, aft.....	P.	2,230	8.3
1	A-47	20-24	Upper platform, ford.	S.	1,240	4.6
3	A-47	20-24	Upper platform, ford.	S.	1,730	6.4
5	A-47	20-24	Upper platform, ford.	S.	1,280	4.8
7	A-47	20-24	Upper platform, ford.	S.	1,590	5.9
2	A-46	20-24	Upper platform, ford.	P.	1,240	4.6
4	A-46	20-24	Upper platform, ford.	P.	1,630	6.1
6	A-46	20-24	Upper platform, ford.	P.	1,280	4.8
8	A-46	20-24	Upper platform, ford.	P.	1,590	5.9
	Total.....				16,040	59.7

Capacities of Reserve Feed-Water Tanks.

	B-94	58½-62	Inner bottom, amid.	P.	5,696	21.2
	B-95	58½-62	Inner bottom, amid.	S.	5,696	21.2
	B-96	62 -65	Inner bottom, amid.	P.	4,863	18.1
	B-97	62 -65	Inner bottom, amid.	S.	4,863	18.1
	B-98	65 -67½	Inner bottom, amid.	P.	4,057	15.1
	B-99	65 -67½	Inner bottom, amid.	S.	4,057	15.1
	Total.....				29,232	108.8

Capacities of Feed and Filter Tanks.

	C-1	80-83	Engine room, aft.....	S.	5,475	20.4
	C-2	80-83	Engine room, aft.....	P.	5,475	20.4
	Total.....				10,950	40.8

Capacities of Trimming Tanks.

	A-1	Stern-3	.....	S. & P.	4,475	16.6
	A-2	3-5	.....	S. & P.	7,467	27.7
	D-12	101-104	.....	S. & P.	14,853	55.1
	D-13	104-Stern	.....	S. & P.	10,162	37.7
	Total.....				36,957	137.1

Recapitulation.

Capacity of fresh-water tanks.....	16,040	59.7
reserve feed-water tanks.....	29,232	108.8
feed and filter tanks.....	10,950	40.8
trimming tanks.....	36,957	137.1
Total, fresh water in tanks.....	93,179	364.4

Capacities of Double-Bottom Compartments.

Compartment.	Frames.	Side.	F.W.	S.W.
			Tons.	*Tons.*
A-93	12 to 15	P. and S.	33.2	34.2
A-94	15 to 18	P. and S.	39.7	40.8
A-95	18 to 21	P. and S.	47.8	49.2
A-96	21 to 24	P. and S.	51.4	52.9
A-97	24 to 27	P. and S.	54.5	56.1
A-98	27 to 30½	P. and S.	66.8	68.7
A-99	30½ to 35	P. and S.	90.4	93.0
B-80	35 to 38	P. and S.	71.2	73.2
B-81	38 to 40½	P. and S.	60.6	62.3
B-82	40½ to 44	P. and S.	87.1	89.6
B-83	44 to 47	P. and S.	75.5	77.7
B-84	47 to 49½	P. and S.	63.9	65.7
B-85	49½ to 53	P. and S.	89.8	92.4
B-86	53 to 56	P. and S.	76.6	78.8
B-87	56 to 58½	P. and S.	63.7	65.5
B-88	58½ to 62	Port	22.7	23.3
B-89	58½ to 62	Starboard	22.7	23.3
B-90	62 to 65	Port	19.2	19.8
B-91	62 to 65	Starboard	19.2	19.8
B-92	65 to 67½	Port	15.9	16.4
B-93	65 to 67½	Starboard	15.9	16.4
C-95	67½ to 71	P. and S.	88.3	90.8
C-96	71 to 74	P. and S.	74.3	76.4
C-97	74 to 77	P. and S.	73.0	75.1
C-98	77 to 80	P. and S.	71.9	74.0
C-99	80 to 83	P. and S.	70.0	72.0
D-96	83 to 86½	P. and S.	68.3	70.3
D-97	86½ to 90	P. and S.	64.7	66.6
D-98	90 to 94	P. and S.	67.7	69.6
D-99	94 to 98	P. and S.	68.2	70.2
Total.....			1,734.2	1,784.1

DESCRIPTION OF DECKS, BRIDGES, ETC.

Masts.—There are two steel masts, the foremast being located at frame 36 and the mainmast at frame 76. They carry the wireless telegraphy system, and each mast has a spar topmast, signal yard, and the following platforms: two on the foremast and three on the mainmast; the upper ones being for searchlights, the lower on the foremast and the middle on the mainmast being range-finder platforms, and the lower on the mainmast carrying searchlights. The conning tower and its support is the foundation for the foremast.

Flying Bridge.—This bridge, built around the foremast, above the forward half of the bridge, is of semi-triangular shape for about fifty feet of the ship's width, with a walk to

the sides of the ship. That portion forward of the mast is the pilot house, in which are the steering wheel, helm and revolution indicators, engine telegraphs, whistle and siren pulls, voice tubes to every part of the ship, etc. Two search-lights are carried, located port and starboard, about 30 feet from the center line of the ship.

Forward Bridge.—This bridge is of irregular shape, being like a rectangle with the corners cut off, is about fifty feet wide by 32 feet long, and is located above the forward end of the upper deck. The conning tower, with entrance to the same, are on this level, and a triangularly-shaped extension forward of the conning tower carries an armored torpedo-directing hatch and station platform. Aft of the conning tower is a chart house and emergency cabin for the commanding officer, the two rooms forming one structure, which is built entirely of brass, the roof of same being used as a platform for the compass. Two 3-pounder and two 3-inch R.F. guns are carried on this bridge.

After Bridge.—This bridge is built around the aftermast and just above the after end of the upper deck. It is about 22 feet long by 50 feet wide, with a walk to the sides of the ship. On it are located six 3-pounder guns. Aft of the mast and flush with the deck of the bridge is an armored torpedo-direction station that extends to the upper deck, the circular foundation for which, between the upper and main decks, being used as the wireless-telegraph station.

Upper Deck.—This deck extends from frame 35 forward to frame 80 aft. All boats are stored on this deck when at sea, except two 30-foot cutters, two 30-foot whale boats, one 30-foot gig whale boat and one 30-foot racing cutter. Two electrically-operated boat cranes, one port and one starboard, are located at frame 64 with their heels stepped on the gun deck; and forward of each, at frame 52, is an electric deck winch. Within the superstructure, under the bridges, hammock berthing has been provided and fresh-water tanks and vegetable lockers located, the space between the bridges being taken up with the general stowage of boats and terminus of exhaust

and ventilator trunks, and fireroom and engine-room hatches. Six 3-inch rapid-fire, four 3-pounder and two 1-pounder guns are located on this deck, as noted in Secondary Battery section of this article. The following boats are carried on this deck: one 40-foot and two 36-foot steam cutters; one 36-foot and three 33-foot launches; three 30-foot cutters, one 30-foot barge; two 20-foot, one 16-foot and one 14-foot dingies; two 14-foot punts.

Total number of boats carried on ship, . . .	23
Total number of men carried in boats exclusive of the punts and racing cutter, ...	807

Main Deck.—The main deck is continuous, extending the full length of the ship. On it are six elliptical, electrically-operated turrets, two of them being 12-inch and four 8-inch, all mounting two guns each. The 12-inch turrets are on the center line of the ship, the center of barbette for the forward one being at frame $28\frac{3}{4}$, and that of the after one at $85\frac{1}{2}$. Between these turrets, port and starboard, at frames $44\frac{1}{2}$ and 69, are the 8-inch turrets, the centers of the barbettes being 28 feet from the center line of the ship, and the superstructure which extends from frames 35 to $79\frac{1}{2}$ cut away so as to give the forward guns a range of 90 degrees forward and the after guns the same range aft. Within the superstructure, between the 8-inch turrets, six 3-inch R.F. guns are located, three being on the port and three on the starboard side. The arrangement on this deck, beginning forward, is as follows: four deck openings for hawse pipes between frames 6 and 9; two anchor davits, one port and one starboard, at frame 9; two 20-foot dingies, harbor position on davits, one port and one starboard, between frames 14 and 18; covered superstructure over anchor-windlass pocket between frames 22 and $24\frac{1}{2}$, the chains passing out through chain pipes in the forward side; two electrically-operated deck winches for anchor work, coaling and general use, one port and one starboard, between frames 25 and 26; two 30-foot cutters, harbor position on davits, sea position on deck, one port and one starboard, be-

tween frames 30 and 36. Aft of the forward bulkhead of the superstructure, at frame 35 to 40, and about 34 feet in width, is the crew's galley; galley issuing room, frames 40 to 41½; No. 1 forward fireroom-exhaust hatch, frames 42½ to 43½; No. 1 boiler hatch, frames 43½ to 46½; general mess pantry, frames 46½ to 48½; No. 1 after fireroom-exhaust hatch, frames 48½ to 50½, with coaling trunks, port and starboard; No. 2 boiler hatch, between frames 52 and 56; officers' galley, frames 56 and 58½; No. 2 fireroom-exhaust hatch, frames 58½ to 59½; No. 3 boiler-hatch, between frames 61 and 65; No. 3 fireroom-exhaust hatch, frames 65 to 66½, with coaling trunks, port and starboard; hammock room, frames 66½ to 70; paymaster's office, starboard, executive officer's office, port, frames 72 to 74; engine hatch, frames 74 to 78; wireless-telegraph room on the center line of the ship, and battalion lockers, port and starboard, frames 78 to 79½, the after bulkhead of the superstructure being located at the latter frame. Aft of the superstructure there are two 30-foot whale boats, one port and one starboard, on davits between frames 78 and 84; two electrically-operated deck winches at frame 90; one 30-foot gig whale boat, starboard, and one 31-foot racing cutter, port, on davits between frames 100 and 105. Outside the superstructure the deck is covered with 3½ by 3½-inch Virginia pine, and inside with linoleum.

Gun Deck.—This deck is also continuous, and the full width of the ship. It has within its space the barbettes for the 12-inch and 8-inch turrets, besides which, in splinter-proof compartments, in sponsons, there are twelve 7-inch R.F. guns. Beginning forward, the arrangement is as follows: Stern to frame No. 8, crew's showers; frame 8 to 14½, crew's wash room; frame 14½ to 28, crew's quarters, with windlass engine amidships, with quarters for master-at-arms on the starboard, and sergeant of marines on the port. Between the 7-inch athwartships armor, at frames No. 28 and No. 86, and between the 12-inch barbettes, the space is divided into twelve splinter-proof compartments, by 1½-inch nickel-steel plate, each compartment containing one 7-inch R.F. gun and all

necessary ammunition hoists and signaling devices. Amidships, between frames 36 and 39, are the bakery, starboard, and bread-mixing rooms, port; No. 1 boiler hatch between frames 41½ and 47, containing drying rooms, entrance hatches, boiler uptakes and ventilator ducts; between frames 46 and 50 is the printing office, starboard, and prison, port; frames 49 to 51 amidships are drying rooms and entrance hatches to middle firerooms; between frames 52 to 56 is No. 2 boiler hatch; between frames 57½ and 60 are drying rooms and entrance hatches to the after firerooms; between frames 61 and 66½ is No. 3 boiler hatch, containing cleaning and drying rooms, entrance hatches to the after firerooms, uptakes and ventilator ducts; amidships, between frames 68 and 78, are the ward-room officers' lavatory and bath, armory and machinery hatch, starboard, and junior officers' lavatory and bath, engineer's office and machinery hatch, port; and outboard, between frames 64 and 68, are the closets for the ward-room officers, starboard, and junior and warrant officers, port; amidships, between frames 79½ and 82, are the ordnance officer's office, starboard, and navigator's office, port. Aft of the 7-inch athwartship armor, at frame 82 to frame 107, the arrangement is as follows: Starboard—Staterooms for the navigator, ordnance and watch officers, captain's bath, stateroom cabin, spare stateroom and bath. Port—Staterooms for senior engineer, senior medical, marine and executive officers, executive officer's bath, captain's office, ward-room mess room. Aft of frame 107 to stern of ship is the captain's after cabin.

Berth Deck.—The arrangement of this deck, beginning forward, is as follows: Frames 3 to 5, oils; frames 5 to 9, starboard—lamps and oils, port—paints; frames 9 to 12, starboard—equipment stores, port—paints; frames 12 to 16, amidships—blower room, starboard—isolation ward and sick-bay lavatory, port—commissary stores and band room; frames 16 to 24, sick bay, containing the operating and dispensary rooms; frames 24 to 35, ice-machine and refrigerator rooms, starboard, and machinists' and petty officers' quarters, port; frames 35 to 40½, starboard—laundry, port—general mess

stores and issuing room; frames 40½ to 72, outboard, starboard and port, are six coal bunkers, inboard of which is a continuous passage, and amidships are the firemen's and servants' wash rooms and the evaporator room; frames 72 to 78, starboard and port, is crew space; and amidships, between frames 73½ and 82, is the general workshop; frames 78 to 94 has junior officers' staterooms and mess rooms, starboard, with the same for the warrant officers, port; frames 94 to 98, junior officers' quarters; frames 98 to 108 has five staterooms for ward-room officers on each side of the ship; frame 108 to stern, ward-room officers' showers.

Protective Deck.—The space under the berth deck on the slope at the ship's sides is divided into numerous compartments and used for various stores, excepting the space between frames 41 and 72, which is a continuation of the berth-deck coal bunkers.

Upper Platform Deck.—The arrangement, beginning forward, is as follows: Stern to frame 5, trimming tank; frames 5 to 9, ordnance stores; frames 9 to 12, starboard—marines' stores, port—medical stores; frames 12 to 16, starboard—dry provisions, port—clothing and small stores; frames 16 to 20, starboard—dry provisions, port—paymaster's issuing room and dry provisions; frames 20 to 24, fresh-water tanks; frames 24 to 35, amidships—four 12-inch shell rooms and 12-inch handling room, starboard—two 12-inch and two 8-inch powder magazines and one 8-inch shell room; port side—similar; frames 35 to 40½, amidships—forward dynamo room; frames 40½ to 67½, divided into six boiler compartments; frames 35 to 46, starboard and port—8-inch ammunition passage and coal; frames 46 to 67½, starboard and port—coal; frames 67½ to 72, amidships—after dynamo room; frames 72 to 83, engine rooms; frames 67½ to 83, starboard and port—8-inch ammunition passage and coal; frames 83 to 94, the arrangement of handling room and magazines is similar to that noted for the forward end of the ship; frames 94 to 98, starboard—navigator, ward room and officers' stores, port—captain and warrant officers' stores; frames 98 to 104, amid-

ships—steering-engine room, starboard and port—electric stores; frames 104 to stern, steering gear.

Lower Platform Deck.—The arrangement, beginning forward, is as follows: Frames 12 to 16, starboard—awnings and linoleums, port—construction stores; frames 16 to 21, torpedo room; frames 21 to 24, amidships—chain lockers, starboard and port—upper hold; frames 24 to 35, amidships—7-inch powder magazines, starboard—7-inch powder magazines, 7-inch shell, 3-inch powder magazine, port—7-inch powder magazine, 7-inch shell, 3-inch ammunition; frames 35 to 40½, amidships—coal; frames 40½ to 67½, amidships—boiler compartments; frames 67½ to 72, coal; frames 72 to 83, engine rooms; frames 40½ to 83, outboard of machinery spaces, ammunition passage, with ten coal bunkers outboard of these on each side of the ship; frames 83 to 84, ammunition passage; frames 84 to 94, amidships—7-inch powder magazines; frames 84 to 94, starboard and port—7-inch shell, 7-inch powder and 3-inch ammunition; frames 94 to 98, torpedo room.

Hold.—The arrangement, beginning forward, is as follows: Stern to frame 5, trimming tank; frames 5 to 18, divided into four compartments on each side of the ship for dry provisions; frames 18 to 21, starboard—naval-defence stores, port—torpedo gear and mine and war heads; frames 21 to 24, amidships—chain lockers, starboard and port, lower hold; frames 24 to 35, amidships—3-inch ammunition magazine, starboard—3-inch powder and saluting powder, 1-pounder and 3-pounder magazines, port—spare magazine, 3-inch field ammunition, and small arms magazines; frames 35 to 67½, boiler spaces and coal, outboard; frames 67½ to 72, coal; frames 72 to 83, engine rooms and coal, outboard; frames 83 to 94, amidships—3-inch ammunition, outboard, starboard and port—machinery stores; frames 94 to 98, starboard—ordnance stores and air compressors, port—war heads and ordnance stores; frames 98 to 101, fresh water; frames 101 to stern, two trimming tanks.

MAIN ENGINES.

There are two main engines, right and left-hand, outboard turning when going ahead, each in a water tight compartment. They are of the vertical, inverted, direct-acting, 4-cylinder, triple-expansion type, designed to develop about 16,500 I.H.P. at about 120 R.P.M., with a steam pressure of 250 pounds at the H.P. cylinder. The arrangement of cylinders, beginning forward, is forward L.P., H.P. and I.P., and after L.P. respectively; the F.L.P. and H.P. and A.L.P. and I.P. being bolted together, thus allowing freedom for expansion between the H.P. and I.P. cylinders in fore-and-aft and vertical planes, while a system of tie rods and braces prevent fore-and-aft and athwartship motion to the engine as a body. The material used for the cylinders is cast iron, fitted with close-grained, hard cast-iron piston and valve-chest liners, the $\frac{3}{4}$-inch space between the casings and piston liners being utilized as the steam jacket, the top and bottom heads of the I.P. and L.P. cylinders also being jacketed. Steam for the jackets is taken from the boiler side of the main-engine throttle valve, and passes successively through the cylinders, spring-reducing valves at the I.P. and L.P. cylinders, allowing a maximum steam pressure of 80 and 60 pounds respectively. Each end of the piston valves is fitted with a packing ring of hard cast iron, made practically solid, being turned larger than the bore of the valve-chest liners, cut obliquely, the abutting ends bolted together and finished to fit the liners. The body of the H.P. valve is cast iron, and those of the I.P. and L.P. valves, steel pipe with cast-iron heads. The H.P. valve stem is connected to the link block, and the I.P. and L.P. stems to a cast-steel crosshead which is connected to the link block, the Stephenson double-bar link motion, with adjustable cut-off, being used. There are five graduations for points of cut-off on each reverse-shaft arm, and are for the ahead motion only.

The cylinders are supported by a framing consisting of twelve forged-steel columns, $5\frac{1}{2}$ inches diameter, flanged top and bottom for bolting to the cylinders and bed plate; each athwartship pair of columns being trussed by X braces, and

stiffened fore and aft by horizontal and diagonal tie rods leading from the middle pairs of columns, thus offering no resistence to cylinder expansion.

The crosshead guides, which are of cast iron, are bolted at the top to facings provided on the cylinders, and at the bottom to cast-steel I-section strongbacks carried by the engine framing. The guides, which are of the slipper type, are hollow for the circulation of water to keep them cool.

The bed plates are of cast steel, made in three sections, flanged and securely bolted together by body-bound bolts. Each bed plate when assembled consists of two longitudinal and six cross girders of I-section, well stiffened by ribs, and furnishes the seatings for the crank-shaft bearings, engine framing and turning engine. Body-bound bolts secure the bed plates to the keelsons, forged-steel washers being fitted at each bolt between the bed plate and the keelson, the space between the washers being filled in with yellow pine.

The crank shaft is arranged with the cranks 90 degrees apart and is in two sections, the forward L.P. and H.P. being in one, and the I.P. and after L.P. on the other, the sequence of cranks being H.P., I.P., F.L.P. and A.L.P.

The upper ends of the connecting rods are forked for the crosshead-pin brasses; these brasses and those of the crank pin being lined with white metal. Crank shafts and connecting rods are high-grade machinery forgings. The pistons are conical shaped, rough machined all over, the material of the H.P. being cast iron, and that of the I.P. and L.P. cast steel, the followers in each case being of the same material. The pistons are bored taper to fit the rods, and have a square counterbore on the under side to suit the shoulder on the piston rods. Pistons are secured to the rods by steel nuts, locked in place by plates bolted to the piston.

The packing rings are of hard cast iron, being practically solid, having been cut obliquely for machining and the ends clamped solidly together; lugs cast on the back of the rings limit the play, the bore being $\frac{1}{8}$ inch greater than the corresponding bore of the piston.

The piston rods are high grade, and the crossheads and slippers class "A" forged steel, the slippers being faced with white metal. The ends of the rods are taperedto fit the pistons and crossheads, and the slippers are bolted to the crossheads.

Engine Framing.—The reversing engine is located inboard and is bolted to the top of the H.P. cylinder. There is an oil-control cylinder of brass connected to the steam cylinder by wrought-steel stanchions which serve as crosshead guides. The reverse shaft is also on the inboard side of the engines, near the top of the engine framing, and is connected to the crosshead of the reversing engine by two forged-steel connecting rods. Differential levers control the valve motion.

The turning engine is located inboard on the after end of the main engine-bed plate. The shaft carries a worm, that engages a worm wheel carried by an inclined shaft, at the lower end of which is another worm that engages the driving worm wheel located on the coupling flanges between the crank and thrust shafts. The second worm is keyed to the inclined shaft by means of a feather, but may be moved vertically to engage or disengage the worm wheel on the crank shaft. A square for a ratchet, for turning by hand, has been provided on the after end of the turning-engine shaft.

MAIN ENGINE DATA.

CYLINDER DATA.

	H.P.	I.P.	L.P.
Diameter, inches	32½	53	61
Stroke, inches	48	48	48
Thickness of body, inches	1⅞	1 7/16	1 5/16
Thickness of liner, inches	1½	1 5/16	1¼
jacket space, inch	⅝	⅝	⅝
valve chest, inches	1 7/16	1¼	1½
Number of studs in cylinder covers	36	48	56
Diameter of studs in cylinder covers, inches	1½	1⅜	1¼
Pitch of studs in cylinder covers, inches	4⅜	4⅜	4
Number of studs in valve-chest covers	18	18	16
Diameter of studs in valve-chest covers, inch	1	1	1
Pitch of studs in valve-chest covers, inches	4½	4½	5¼
Number of tap bolts in piston liners	24	36	36
Diameter of tap bolts in piston liners, inch	⅞	⅞	⅞
Pitch of tap bolts in piston liners, inch	4½	4½	5¼

	H.P.	I.P.	L.P.
Number of piston valves	1	2	2
Diameter of piston valves, top	17½	19	19
bottom	17 7/16	18 1/16	18 1/16
balance pistons, inches	9	9	9
Port area through valve-chest liner, top, inches	150.5	166.92	190.83
bottom, inches	158.7	166.22	190.04
Valve travel, inches	10	10	10

VALVE DATA FOR PORT ENGINE.

Steam lead, top, linear, inches	½	⅝	1 1/16
bottom, linear, inches	9/16	1⅛	1⅛
Full gear cut-off, top, per cent	80.8	76.5	74.0
bottom, per cent	72.2	69.0	63.7
Earliest cut-off, top, per cent	56.0	51.5	51.7
bottom, per cent	44.5	43.7	39.9
Diameter of valve stem through gland, inches	2⅞	2⅞	2⅞
valve, inches	1⅞	1⅞	1⅞

	H.P. and I.P.	L.P.
Diameter of piston rods, inches	7½	7½
axial hole, inches	1½	3 4⅜
crosshead pins, inches	9½	9½
Length of crosshead pins, each, inches	9½	9½
slippers, inches	27½	27½
Width of crosshead slippers, inches	21	21
Length of backing surface, inches	27½	27½
Width of backing surface, each, inches	7⅞	7⅞
Length of connecting rod, center to center, inches	96	96
Diameter of connecting rod, top, inches	7	7
bottom, inches	8	8
hole, body, inches	1½	3⅜ 4⅞
crosshead end, inches	1½	1½ 1½
pin bolts, inches	3	3
Number of crosshead-pin bolts	4	4
Diameter of crank-pin bolts, inches	4	4
Number of crank-pin bolts	2	2
Diameter of crank pins, inches	17½	17½
Length of crank pins, inches	19	19
Diameter of hole in crank pins, inches	10	10
shaft, inches	9½	
crank shaft, inches	17½	
journals, inches	16½	
coupling flanges, inches	29½	
Width of crank-shaft coupling flanges, inches	3½	
Number of coupling bolts	16	
Diameter of coupling bolts, inches	3½	
pitch circle, inches	23½	

Width of crank webs, inches	19
Thickness of crank webs, inches	10½
Number of main bearings	6
Length of main bearings, H.P., inches	19½
I.P., inches	19½ and 10½
forward L.P., inches	19½
after L.P., inches	10¾ and 19½
Diameter of main-bearing bolts, inches	2¾
Number of main-bearing bolts per bearing	4
Number of packing rings in pistons	1
Width of packing rings in pistons, inches	2½
Thickness of packing rings in pistons, inch	1
Number of follower studs in H.P. pistons	10
I.P. pistons	16
L.P. pistons	17
Diameter of follower studs, inches	1½

REVERSING ENGINE.

Diameter of steam cylinder, inches	13
oil cylinder, inches	7
steam piston rod, inches	2⅞
oil piston rod, inches	2⅞
Stroke, inches	19⅝
Length of connecting rods, feet and inches	3-05⅛
reverse levers, inches	16½
Diameter of reverse shaft, inches	7 and 6
bearings, inches	6½
Length of reverse-shaft bearings, inches	8½

TURNING ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	6
piston rods, inches	1½
valves, inches	3½
Stroke, inches	7
Valve travel, inches	1½
Crank angle, degrees	90
Area of crosshead slipper, inches	16⅝
Diameter of crosshead pin, inches	1¾
Length of crosshead pin, inches	2¾
Diameter of crank pin, inches	2½
Length of crank pin, inches	2½
Diameter of crank shaft, inches	2½
Length of crank-shaft bearings, inches	4 and 3
Diameter of pitch circle of worm, inches	6⅛
wheel, inches	17⅜
Number of teeth	36

Diameter of pitch circle of vertical worm, inches	9
worm wheel, inches	64 1/2
Number of teeth	60
Diameter of vertical shaft, inches	4 1/2
Area of step bearing, square inches	38 1/2

ASSISTANT CYLINDERS.

An interesting and important feature of the propelling machinery is the Lovekin Improved Assistant Cylinders, applied to the valve gear of the main engines, this ship being the second in the United States Navy to be equipped with this simple and yet most effective device for overcoming the inertia primarily the cause of considerable wear upon the heavy valve gear working at high speeds with the resulting item of expense for repairs; and an unnecessary amount of vibration so annoying at any time and especially so on shipboard.

The highly satisfactory results obtained were anticipated, as the cylinders had proven their usefulness upon the armored cruiser *Washington*, and were the subject of an exhaustive article at that time; but an elaboration of diagrams and calculations were not necessary to prove to the practical man that the device was effective, as the smooth-running valve gear was sufficient evidence of that.

For results obtained upon the *Kansas*, there is given here (Plates III and IV) an analytical diagram that briefly illustrates the work done by the assistant cylinder on the port H.P. valve gear. The ordinates of the lines A B and B C, measured from the base line, give the inertia resistance to be overcome by the cylinder at the various stages of the valve travel. The compression and expansion curves are exact reproductions of cards taken on the trial. The difference in the lengths of the ordinates of the upper and lower curves gives the ordinate of the resulting curve; and the shaded area between the inertia lines and the resultant effort curve represent the unbalanced work which has to be done by the eccentrics and is fully explained in the following articles, the

H P

Top

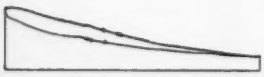
Bottom

I P Out

Top

Botto

L P Aft Outbd.



Top

Bottom

L P Aft Inbd.

Top

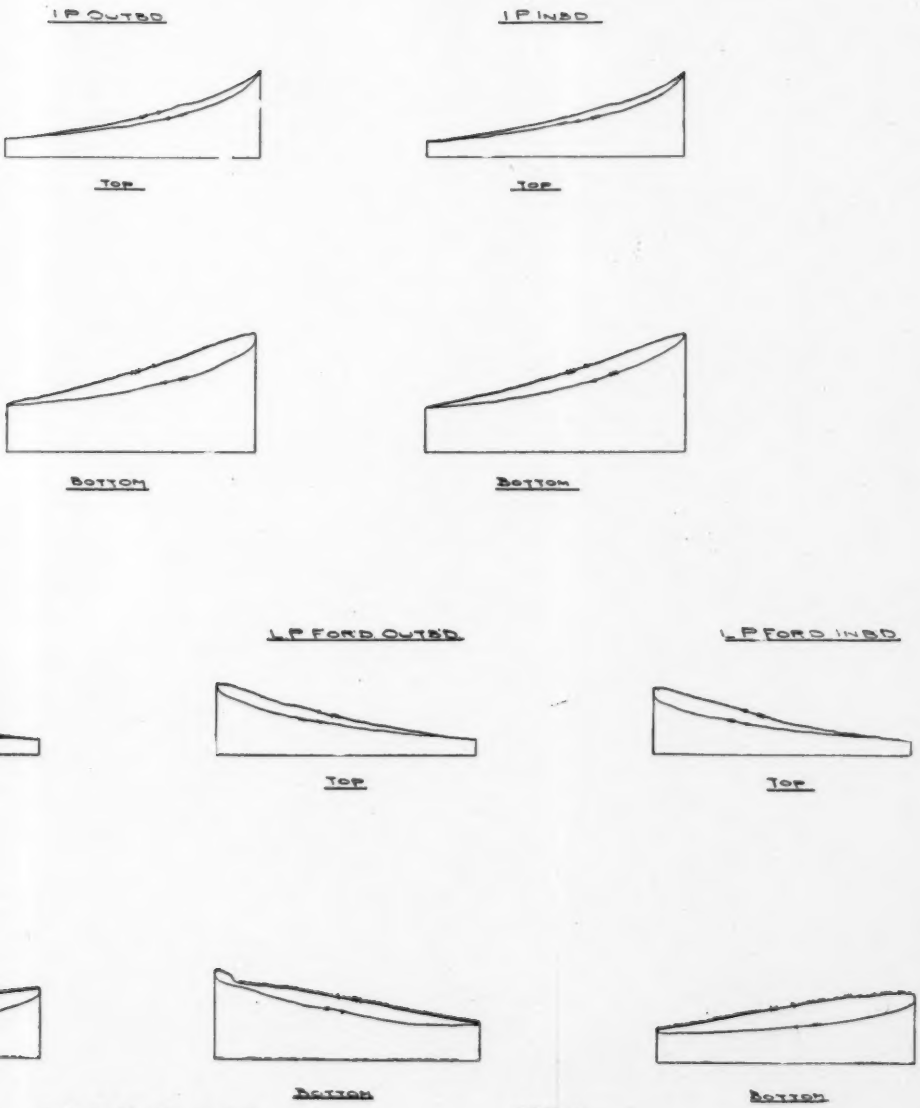
Bottom

Scale 1" = 2.4".

Scale of sprin

U. S. S. "Kansa

Indicator cards taken from the Lovekin Im

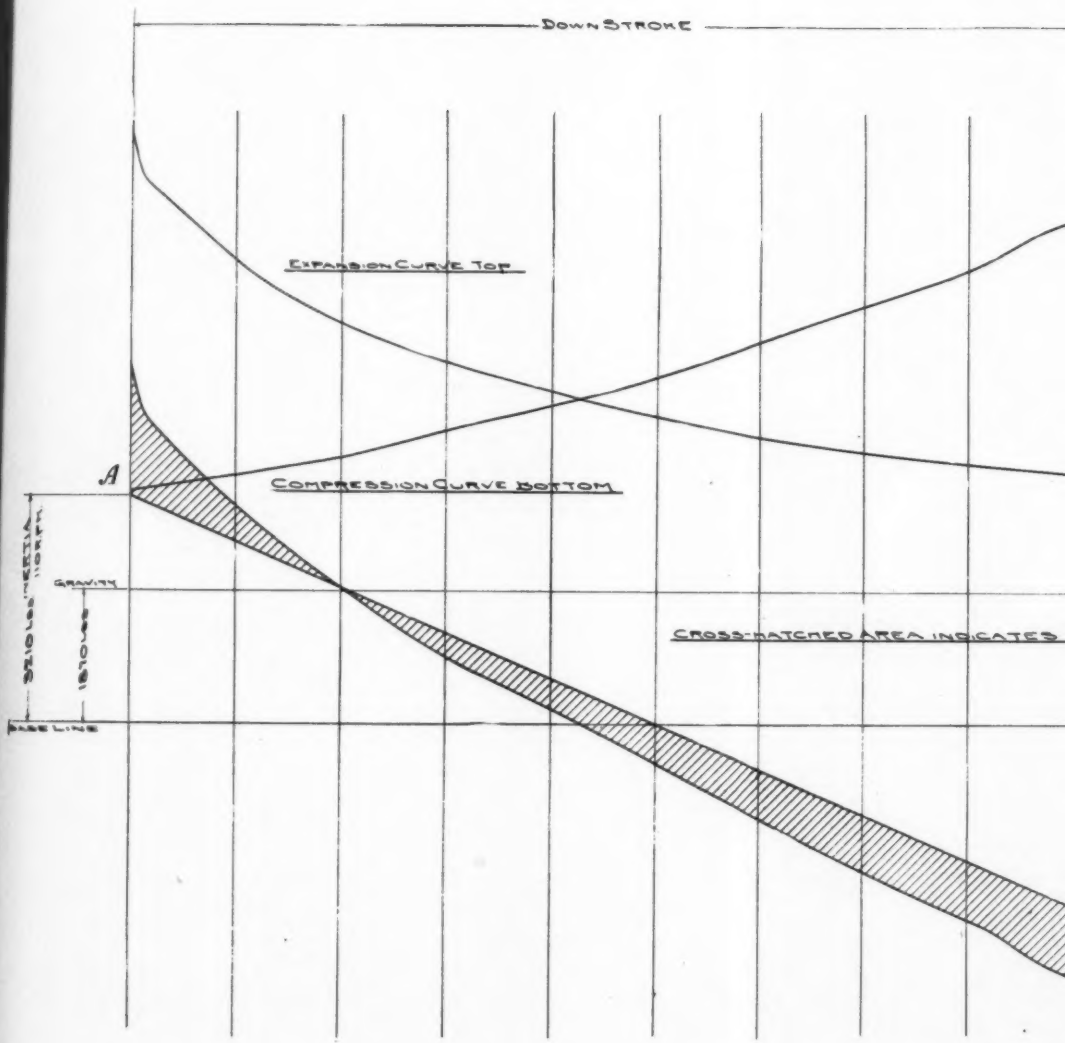


Scale of springs 1″ = 149 pounds.

R.P.M. 110.

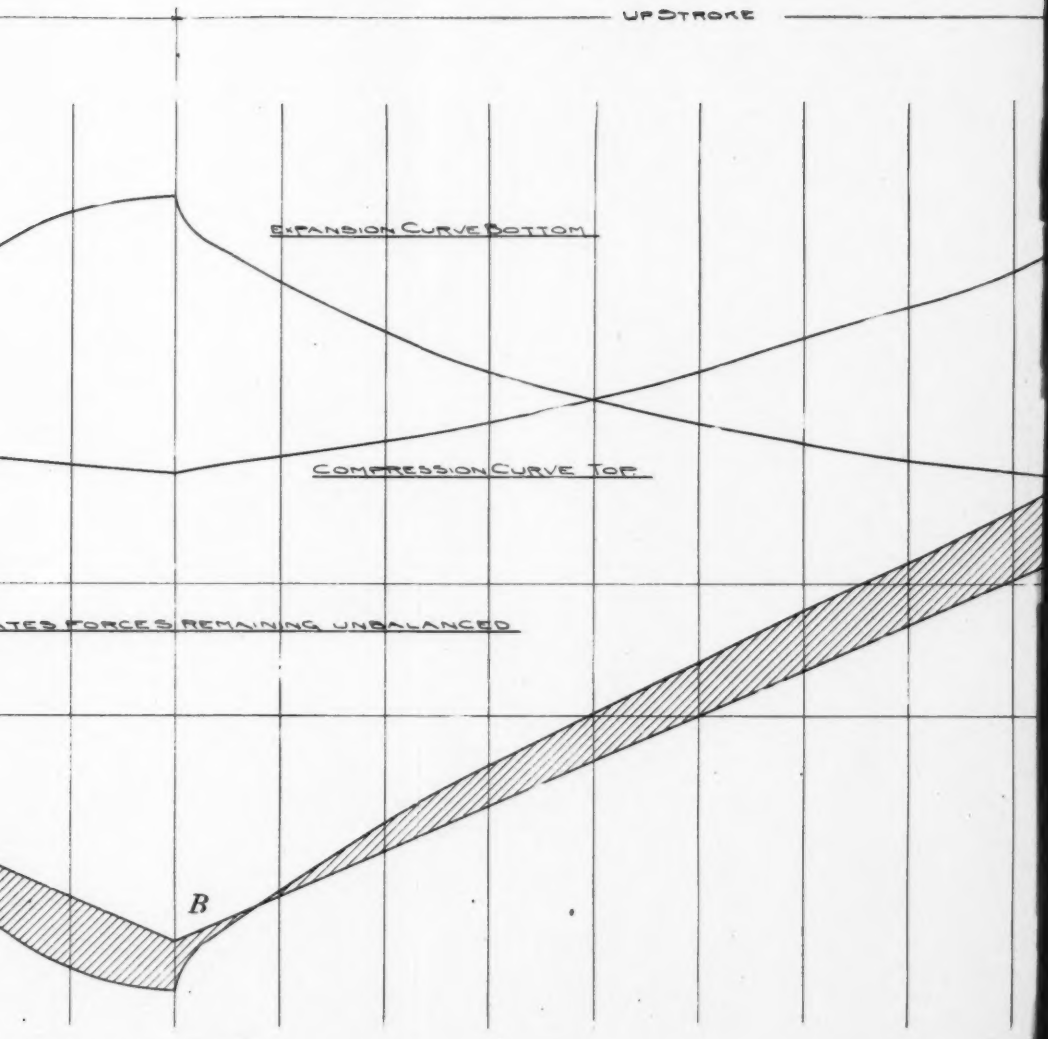
S. S. "KANSAS."—**Plate III.**

he Lovekin Improved Assistant Cylinders on the *Kansas*.



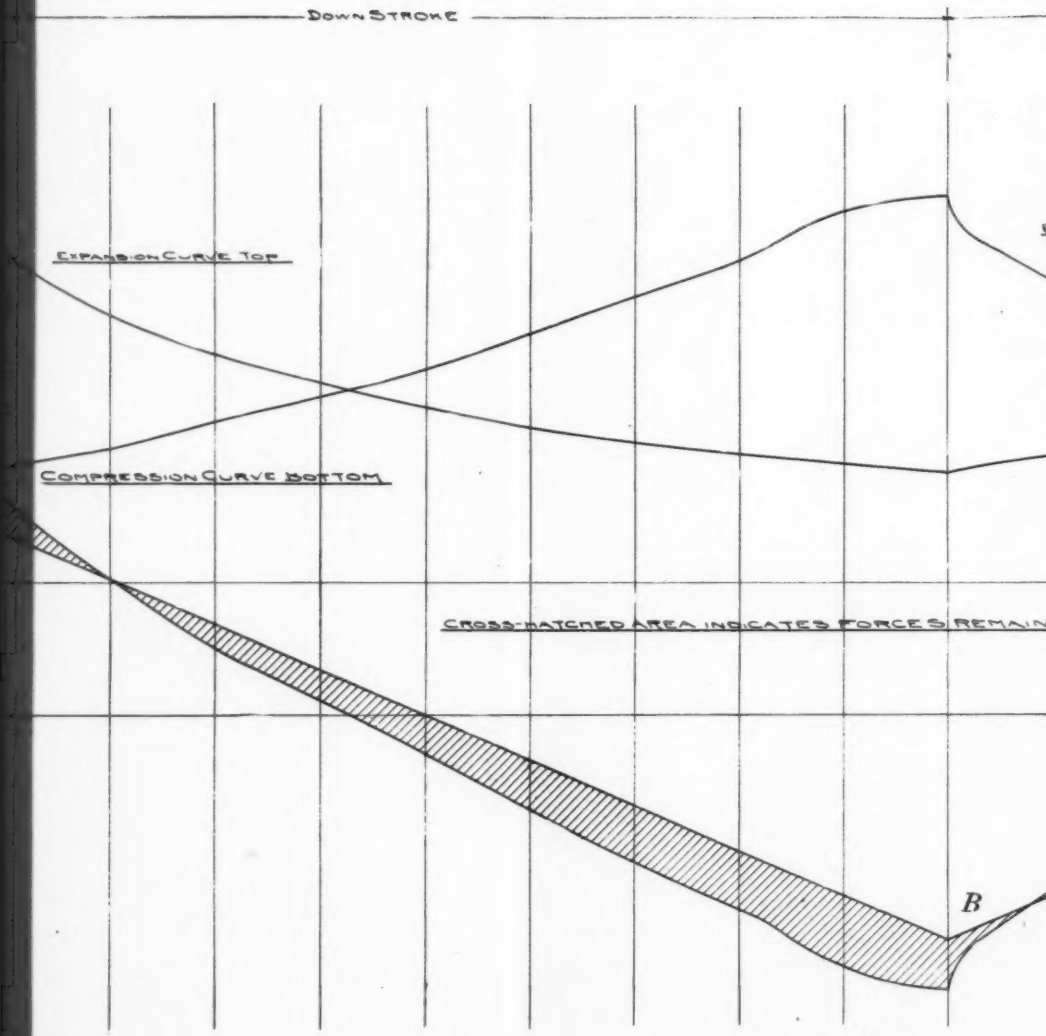
U. S. S. "I

Revolutions per minute = 110.



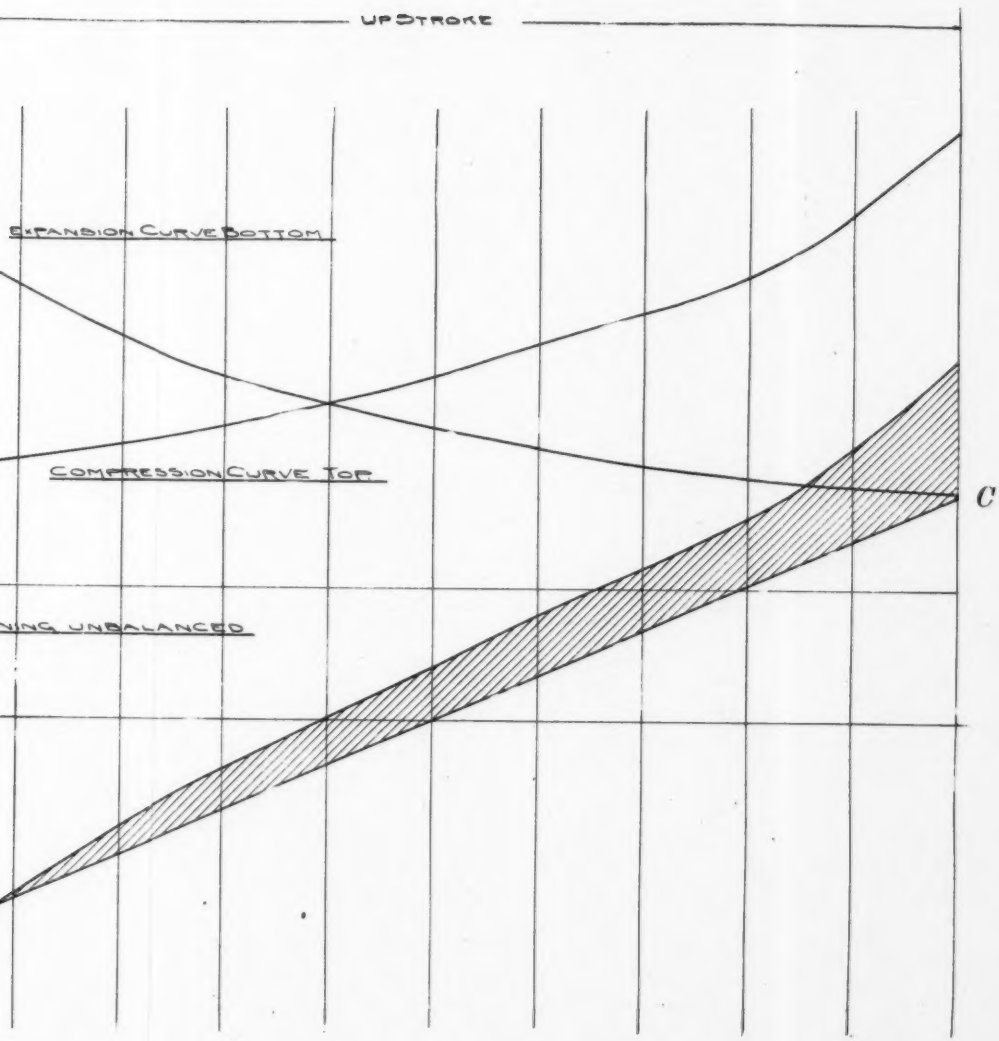
S. "KANSAS."—**Plate IV.**

= 110. Scale of forces 1 inch = 2,410 pounds.



U. S. S. "KANSAS."—**Plate IV.**

Revolutions per minute = 110. Scale of forces 1 in



nch = 2,410 pounds.

limited scope of this article not permitting an exhaustive analysis of the subject. See Plates Nos. III and IV.

Interested readers will be well repaid by consulting the following references: "Balance Cylinders, Theoretically and Practically Considered," "Performance of the Assistant Cylinders of the *Washington*," and JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XVI, No. 3; XVII, No. 3; XVIII, No. 3, and "The Balancing of Valve Gears," "Journal of the Society of Naval Architects and Marine Engineers," Nov., 1902.

SHAFTING AND THRUST BEARING.

The shafting is arranged in four sections, as follows: Thrust, line, stern-tube and propeller, all of class A, forged steel. The line and stern-tube shafts are coupled as follows: A forged-steel collar fits on the end of the stern-tube shaft, secured, to prevent turning, by three keys, and into a circumferential groove in the end of the shaft is fitted a ring made in halves, the coupling bolts passing through the steel collar, the half rings and the flange on the thrust shaft. The stern tube and propeller shafts are coupled as follows: Over the tapered ends of the shafts is fitted a long sleeve bored with a double taper to fit that of shafts and secured to each by two keys and a taper cotter.

The thrust bearing is of the horseshoe type, with a steady bearing at each end, fitted with glands to prevent the escape of oil. The body is of cast iron, so shaped as to form an oil reservoir, and the horseshoes are of cast steel, made hollow for the oil and water service. They are faced with white metal and have oil grooves on the bearing surface.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	15¼ and 15¾
hole, inches	9½
Number of thrust-shaft collars	12
Outside diameter of thrust-shaft collars, inches	24½
Width of thrust-shaft collars, inches	2
space of thrust-shaft collars, inches	4
Length of thrust-shaft bearings, inches	18

Number of thrust shoes	12
Diameter of thrust side rods, inches	3⅜
line shafting, inches	15½
stern-tube shafting, inches	16½
shaft hole, inches	9½
Length of stern-tube shaft forward bearing, inches	54
after bearing, inches	54½
Diameter of propeller shaft, inches	16½
hole, inches	9½
Length of strut bearing, inches	51
Diameter of thrust-shaft coupling flange, inches	29½
Thickness of thrust-shaft coupling flange, inches	3½
Number of thrust-shaft coupling bolts	8
Diameter of thrust-shaft coupling bolts, inches	3
pitch circle, inches	23½
Length of collar on stern-tube shaft, inches	11⅝
Diameter of collar on stern-tube shaft (inside), inches	16½
Number of keys	4
Length of keys, inches	11¼
Width of keys, inches	2½
Depth of keys, inches	1⅝
Thickness of halved ring, inches	2
Inside diameter of halved ring, inches	14½
Length of outboard coupling sleeve, inches	67½
Outside diameter of outboard coupling sleeve, inches	21½
Width of longitudinal keys, inches	2½
Depth of longitudinal keys, inches	1½
Width of cotter keys, top, inches	7½
bottom, inches	6½
Thickness of cotter keys, inches	2

PROPELLERS.

The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a composition nut on the end of the shaft, the nut being threaded the reverse of the direction of the propeller. The nut is locked in place and covered with a composition cap bolted watertight to the hub. The blades fit into tapered recesses in the hub, being held in place by rolled manganese tap bolts, composition chocks being fitted around the bodies of the bolts to keep the blades from shifting after being set to the desired pitch. The bolt holes in the hub are oval in shape and allow a pitch adjustment of one foot either way. The pro-

pellers were accurately balanced, and each blade was made a true surface on both faces, by machining, grinding and polishing. The material for blades and hubs is manganese-bronze.

DATA FOR ONE PROPELLER.

Number of blades	3
Diameter of propeller, feet and inches	17-04
Pitch for official trial, feet and inches	17-06
maximum, feet and inches	18-06
minimum, feet and inches	16-06
Ratio of pitch to diameter	1.009
Helicoidal area, square feet	96.0
Projected area, square feet	82.5
Disc are, square feet	235.97
Ratio projected to disc area	.35
Area of immersed midship section, square feet	1,811.0
Ratio disc to immersed midship-section area	.26
projected to immersed midship-section area	.091
Number of tap bolts for blades	27
Diameter of tap bolts for blades, inches	3½
Length of tap bolts for blades under head, inches	6½
Width of longitudinal key, inches	3½
Thickness of longitudinal key, inches	2½
Length of longitudinal key, inches	36
Diameter of shaft-nut thread, inches	9½
Length of shaft-nut thread, inches	9
Number of threads per inch	4
Immersion of tip of blade, inches	74⅞
Tip of blade above keel, inches	10⅛

DATA FROM OFFICIAL TRIAL.

	Port.	*Starboard.*
Average revolutions per minute on course	120.52	122.12
Slip, per cent. of its own speed	14.29	15.43

BOILERS.

The boilers on this ship are of the Babcock & Wilcox type, and were designed, built and installed by this well known concern. The battery consists of sixteen boilers placed two each in eight watertight compartments, all compartments being symmetrically arranged.

Each boiler has one drum extending the full width of the boiler and is located within the casings, so that about one-

fourth of the shell is in contact with the gases at the top of the combustion chamber. Immediately below the drum are the front headers, and at the back of the boilers are the back headers, connected by tubes set at an angle of 15 degrees.

The weight of the drum and front headers, etc., is carried by the furnace front, and that of the back headers by a light structural-steel girder, that also forms a portion of the casing.

This is protected from the fire by a bridge wall of brick extending the full width of the furnaces. There are two of these of equal size, separated by a wall of rectangular water legs that extends from the lower row of tubes to a point just below the surface of the grate. The side walls are similarly constructed, and all are connected to a mud drum carried in the boiler front immediately below and connected to the front headers, thereby providing a constant circulation of water and affording excellent means for draining and blowing off.

The boiler casings are made of steel plates and shapes, fire brick and block magnesia being used for insulation. Front and back at the header space they are in the form of doors of such size that they may be easily removed for access to the inside of the tubes, and the sides are removable in sections for access to the baffle plates and for cleaning soot from the tubes.

Each furnace is provided with three furnace and two ash-pit doors, and the grate bars are of the ordinary common type, of such size and shape as to be easily handled by one man.

Boiler Drums.—The boiler drums are of open-hearth steel plate, having two seams which are butt-strapped inside and out. The heads are convex, the radius of which is equal to the inside diameter of the drums, and in each is a flanged manhole shaped from the same plate. All butt straps were formed to the proper curvature of the drum under hydraulic pressure, the rivet holes being punched about $\frac{1}{4}$ inch smaller than the diameter of the rivets used, and then drilled out full size after the plates were rolled and assembled. After drill-

ing all burrs were cleaned off, the plates reassembled, with turned bolts holding the parts in place for riveting. Wrought-steel pads securely riveted to the drum shell are provided for attaching all valves and fittings.

The following external fittings are provided for each drum :

One 5-inch hand- and steam-closing boiler stop valve.
One twin-spring safety valve, each valve 4½ inches in diameter.
Two 2½-inch stop valves for main and auxiliary feed.
Two 2½-inch check valves for main and auxiliary feed.
Two Klinger reflex water gauges with automatic fittings.
One 1½-inch surface-blow valve.
One ½-inch sentinel valve.
One ½-inch air valve.
One ¼-inch steam-gauge connection.
Three ⅜-inch try cocks.

The following internal fittings are provided :

One 6-inch brass dry pipe.
One 2½-inch main feed pipe.
One 2½-inch auxiliary feed pipe.
One 1½-inch pipe and scum pan for surface blow.
Zinc protectors and baskets.

Headers.—The headers are of forged steel, forged in one piece into a rectangular section, having male and female sinuous side for nesting when assembled. The front side is provided with numerous handholes for access to the generating tubes. The covers for these handholes are of pressed steel held in place by one stud, nut and dog. All holes for the generating tubes and connecting nipples between headers and drums are bored and reamed the size of tubing used.

Tubes.—The generating tubes are straight and made of seamless cold-drawn steel. These and the header and drum nipples, which are of the same material, are rolled in place by the B. & W. patent expander. Only one nipple showed any sign of leak on the trial.

DATA FOR BOILERS.

Number of boilers	12
Length of drum, feet and inches	15-03
Inside diameter of drum, inches	42
Thickness of drum plate, inches	1 1/2
Heating surface per header and tubes, square feet	175.8
Total heating surface for one boiler, square feet	4,396
grate surface for one boiler, square feet	91.56
H.S. for all boilers, square feet	52,752
G.S. for all boilers, square feet	1,000
Ratio H.S. to G.S	47.95
Number of boilers per smoke pipe	4
Height of smoke pipe above grate, feet and inches	92-05
Area through smoke pipe, square feet	53.46
Ratio G.S. to smoke-pipe area	6.85
Size of generating tubes, inches	2 and 4
Total number of headers, per boiler	25
Thickness of 2-inch generating tubes, B.W.G	8
4-inch generating tubes, B.W.G	25
Exposed length of generating tubes, feet and inches	9-0
Number of 2-inch generating tubes per boiler	786
4-inch generating tubes per boiler	25
Inclination of tubes, degrees	15
Height of boiler, feet and inches	13-06 1/2
Length of boiler, feet and inches	10-01
Width of boiler, feet and inches	16-04 1/2
Weight of one boiler complete, dry, pounds	66,791
water in one boiler, pounds	15,208
one boiler complete, wet, pounds	81,999
Total weight of boilers, dry, pounds	1,068,656
water in boilers, pounds	243,328
boilers, wet, pounds	1,311,984
Weight of boiler, dry, per cubic foot, pounds	29.89
wet, per cubic foot, pounds	36.70
Ratio of weight of boilers dry to boilers wet	.814

MAIN STEAM LINE.

The main steam pipes are arranged symmetrically in two systems, one on each side of the center-line bulkhead. They are united in the engine rooms by an 8-inch cross-connecting pipe, with a valve worked from both sides of the center-line bulkhead. The main steam pipes are supplied directly from the auxiliary steam pipes by a 7-inch globe valve at the after end of each boiler room, consequently there are no main steam lines in the forward boiler compartments. From the forward

and middle compartments the branches unite into a 9-inch pipe, which is increased to 11 inches at the junction of the connection from the after compartment. This diameter is carried aft through the dynamo rooms to the engine rooms, passing through the main-engine separators which are located on the center-line bulkhead in the forward end of each engine room, directly under the protective deck. The same diameter is carried from the separators to the main-engine throttle, and between these points a main stop or cut-out valve is located at the separator, and the expansion in this line is taken care of by a balanced expansion joint anchored to the forward engine-room bulkhead. From a 6-inch nozzle on the separators a cross connection is led between the main and auxiliary steam lines. At the separators nozzles are also provided for 5-inch branches to the receivers, and 5-inch bleeder connection to main condensers.

Globe stop or cut-out valves are placed just forward of each junction, and all valves in the main steam line have by-pass valves attached to relieve them when jammed on their seats.

The pipes are of seamless-drawn steel tubing with rolled-steel flanges. The pressure on the piping is 265 pounds per square inch, and the thicknesses of the pipes were determined by the following formula:

$$T = \frac{P \times D}{10{,}000} + \tfrac{1}{8} \text{ inch};$$

where

P = pressure above atmosphere, in pounds, per sq. inch.
D = inside diameter of pipe, in inches.
T = thickness, in inches.

The expansion of the main steam pipes in the firerooms and dynamo rooms is taken care of by an expansion joint in each compartment, and where pipe is bolted to athwartship bulkheads proper allowance has been made for expansion by making bolt holes oval.

Where condensed steam is apt to accumulate, drain valves

are fitted and pipes led to automatic float traps which discharge into condensers or feed tanks.

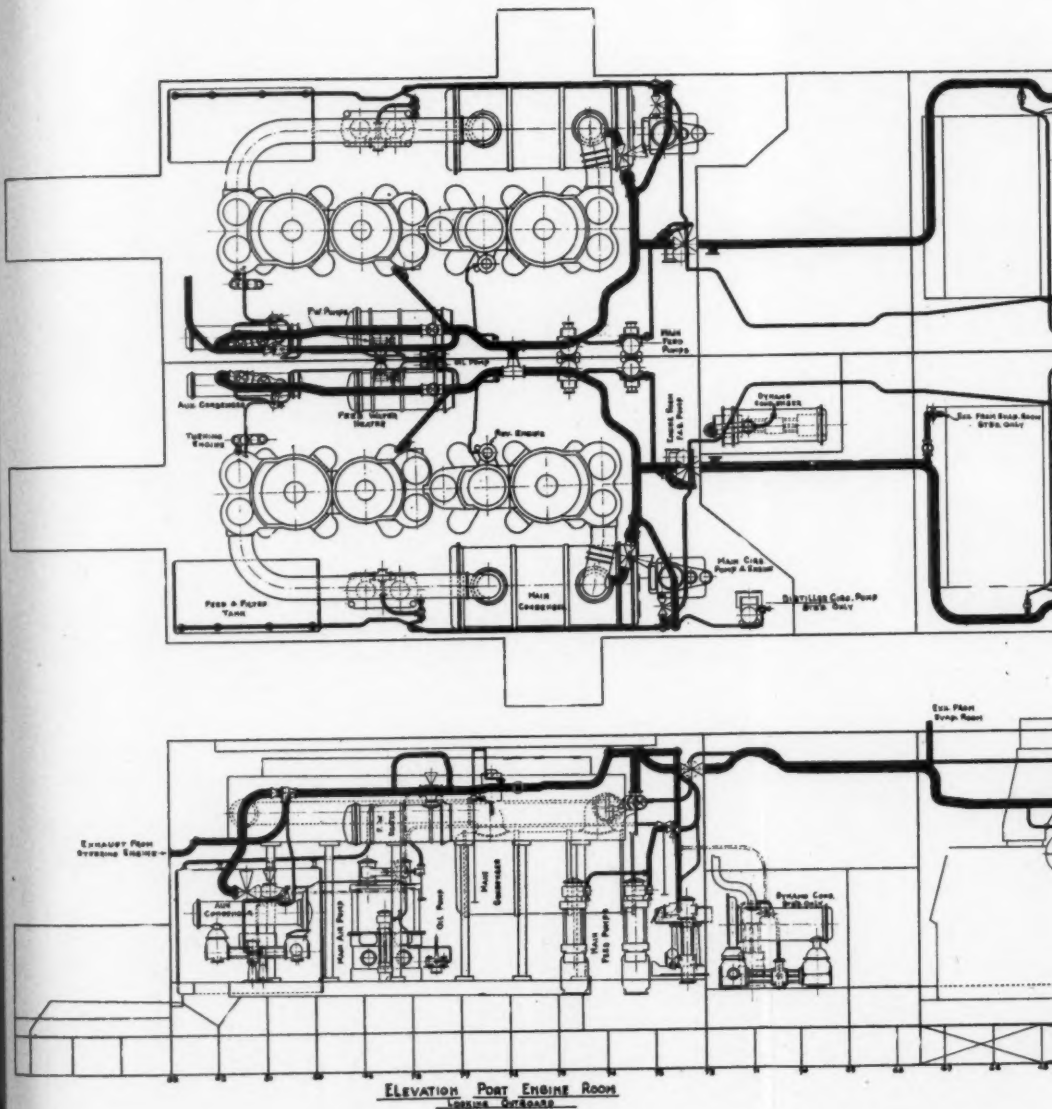
The piping is lagged with sectional magnesia covering with canvas sewed on and well painted and lagged with Russia sheet iron.

AUXILIARY STEAM LINES.

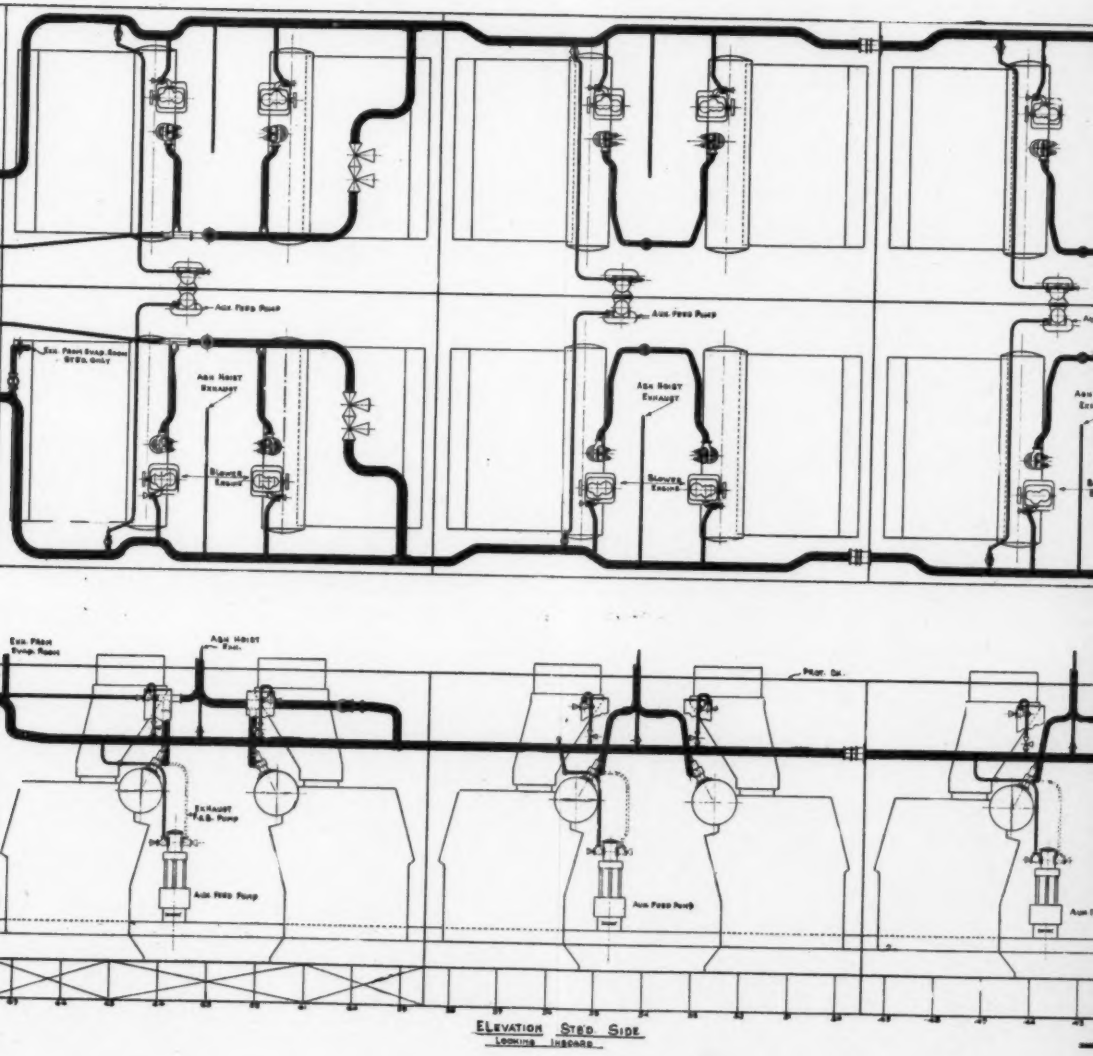
The auziliary steam lines are arranged in two systems, extending fore and aft through the boiler compartments, one on each side of the center-line bulkhead, directly under the main steam line. Each system is supplied directly from the boilers by connecting with the main stop valves of all boilers on its own side of ship through 5-inch branches. From the connection on the forward boiler in the forward boiler compartment the size of the pipe is carried 7 inches to the after end of the after boiler compartment, from which a 4-inch branch is taken on the port side to supply steam to the evaporating plant. Aft of this the pipe is reduced to 6½ inches diameter, and carried through the dynamo room to the engine rooms, with stop or cut-out valves on the forward dynamo-room bulkhead. Between these valves and the forward engine-room bulkhead are 4-inch branches for steam to after dynamo room, and aft of these branches the pipe is reduced to 6 inches diameter to supply steam to the auxiliaries in the engine rooms.

With a valve at the forward dynamo-room bulkhead the auxiliary steam lines are carried along the inboard sides of the engine rooms, and the two systems are united by a 6-inch connection passing through the center-line bulkhead with a valve at the bulkhead.

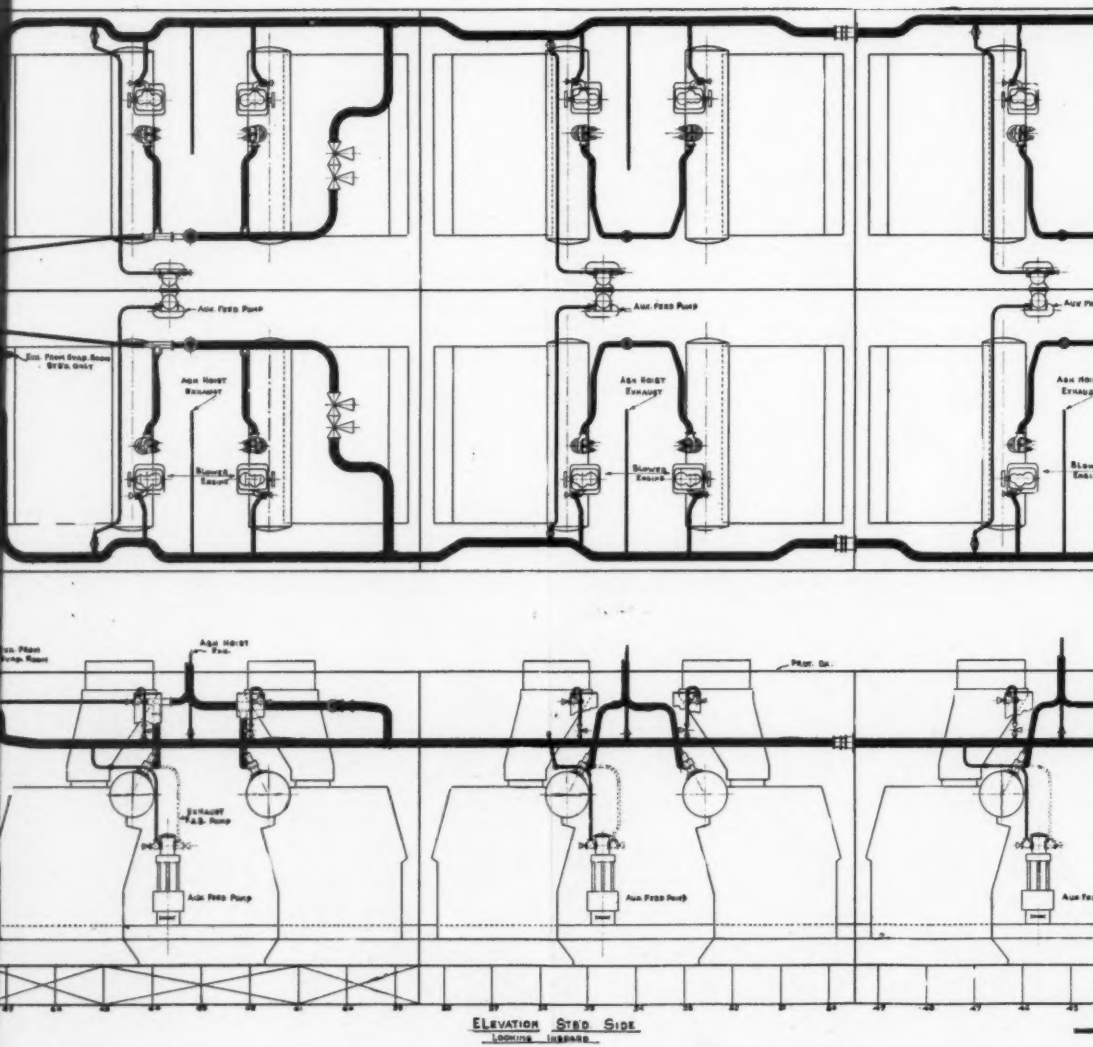
Steam to the steering engine is supplied by a 4½-inch branch taken off of the auxiliary steam line in the engine room on the starboard side. The distributing castings in the engine room also supply steam to main feed pumps, main air pumps, main circulating-pump engines, engine-room fire and bilge pumps, distiller circulating pumps, auxiliary air and circulating pumps, fresh-water pumps, oil pumps, turning and reversing engines, cylinder jackets, after heating system and



ELEVATION PORT ENGINE ROOM
LOOKING OUTBOARD

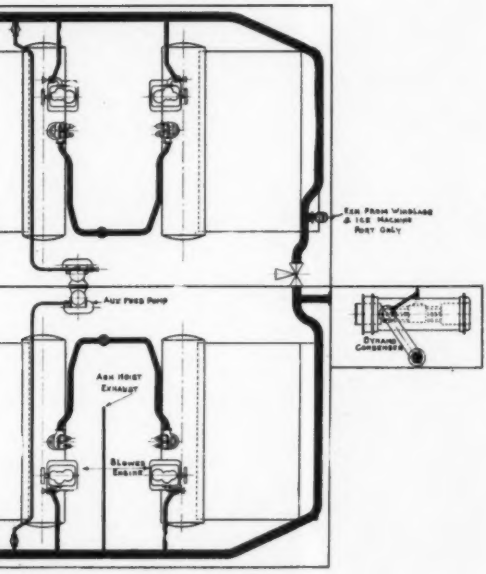


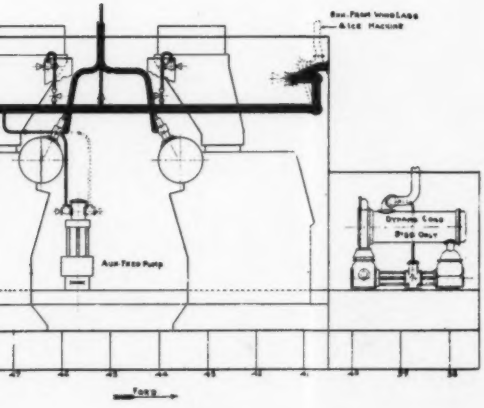
U. S. S. "KANSAS."—**Plate V.**

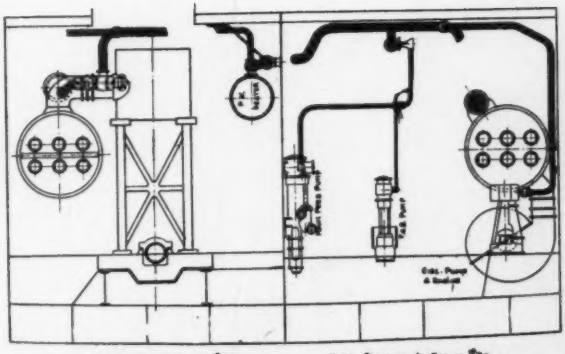


ELEVATION STBD SIDE
LOOKING INBOARD

U. S. S. "KANSAS."—**Plate V.**

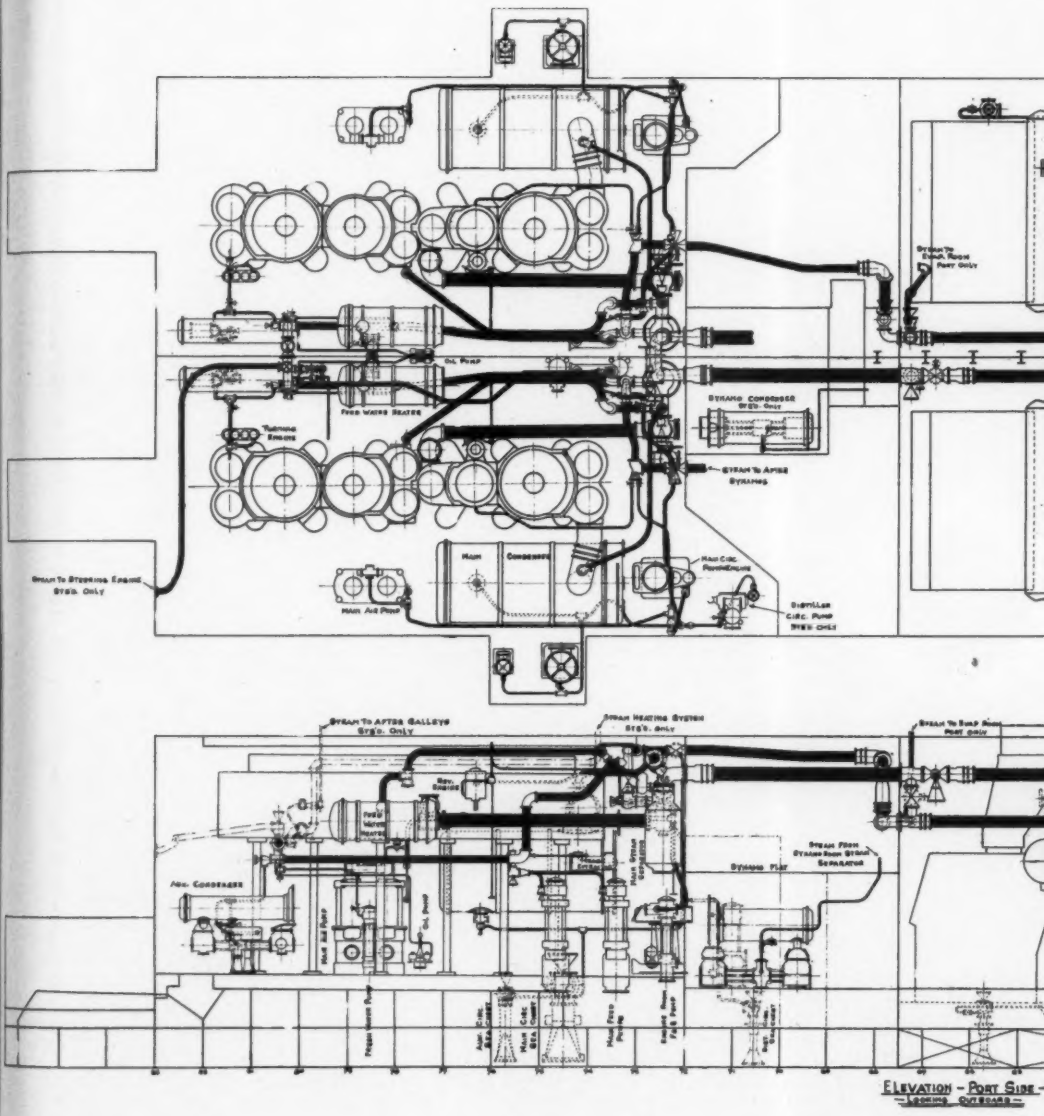




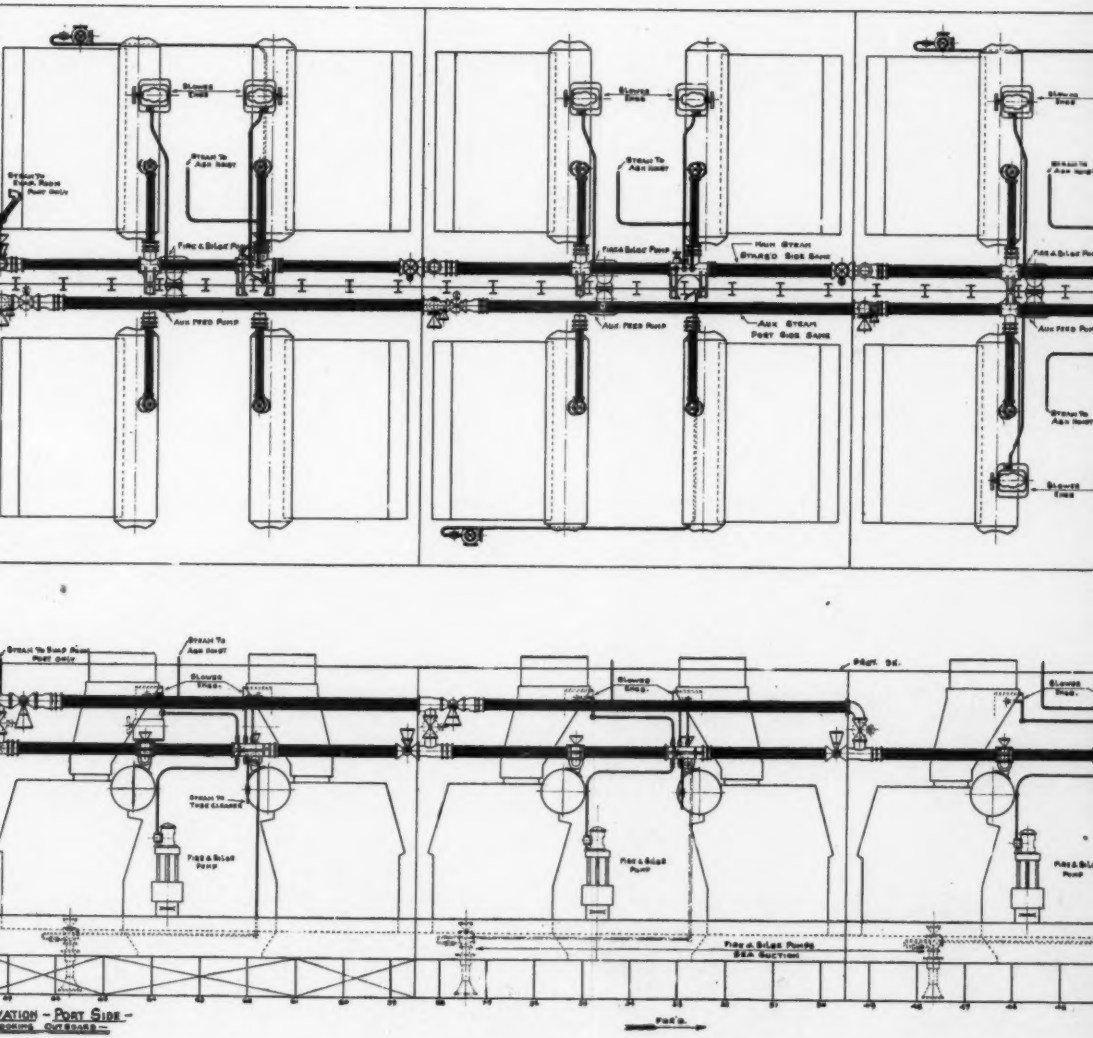


Half - Section At Frame #78
Looking Aft.

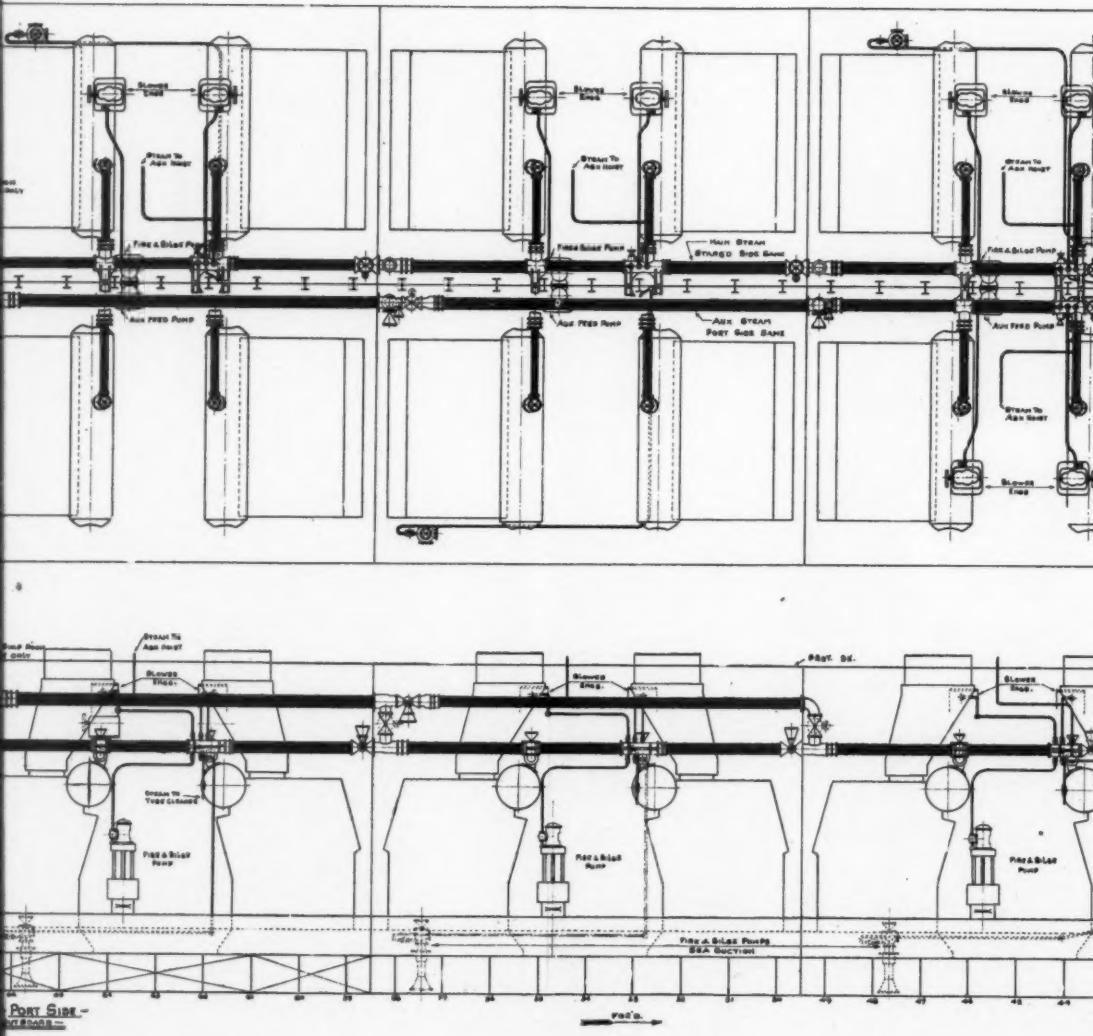
Half Section At Frame #78
Looking Aft.



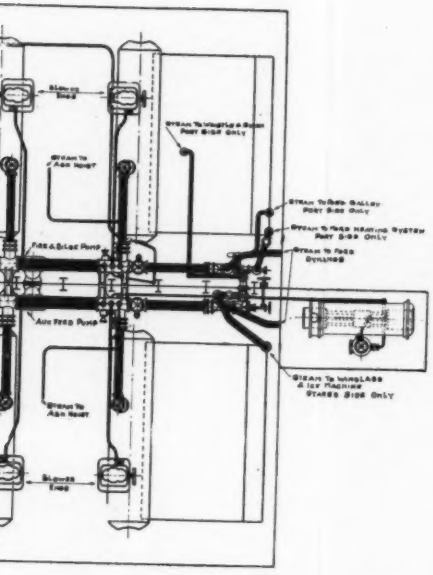
ELEVATION – PORT SIDE
– LOOKING OUTBOARD –

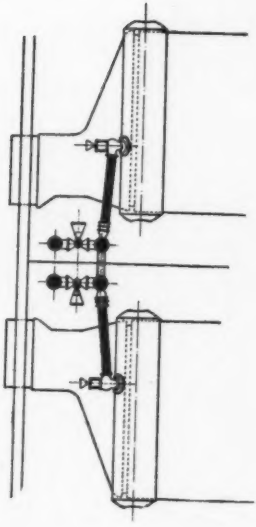

Blowoff Ends
Steam To Ash Hoist
Fire & Bilge Pump
Aux Feed Pump
Main Steam Starbd Side Gear
Aux Steam Port Side Gear
Steam To Tube Cleaner
Fire & Bilge Pumps Sea Suction

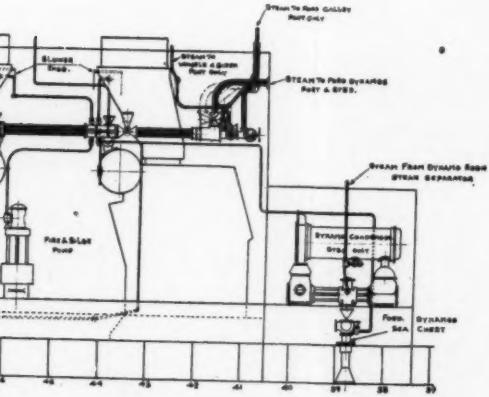
U. S. S. "KANSAS."——**Plate VI.**

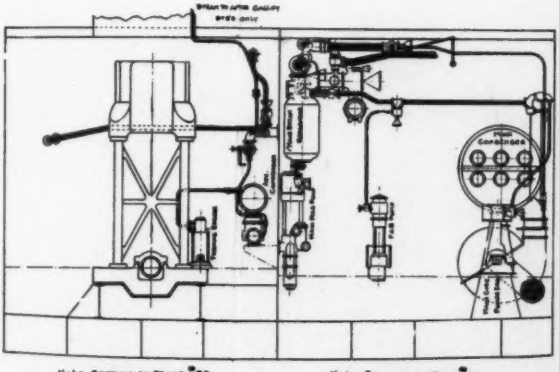


U. S. S. "KANSAS."——**Plate VI.**


Steam To Whistle & Siren Port Side Only
Steam To Fore Galley Port Side Only
Steam To Fore Heating System Port Side Only
Steam To Fore Dynamos
Fire & Bilge Pump
Air Feed Pump
Steam To Whistle & Ice Machine Stokes Side Only
Steam To Aft Heat
Steam To Fore Heat




Steam To Fore Galley Port Only
Steam To Whistle & Siren Port Only
Steam To Fore Dynamos Port & Stbd.
Steam From Dynamo Fore Steam Separator
Fire & Bilge Pump
Dynamo Condenser
Fore Dynamo Sea Chest



Half Section at Frame "80" Looking Aft.

Half Section at Frame "78" Looking Aft.

galley, and connections for blowing out sea chests and boiling out condensers, feed-water heaters and grease extractors.

In the forward fireroom just forward of the 5-inch connection to the forward boiler the systems are reduced to 6 inches and fitted with cut-out valves. These 6-inch connections extend to the forward end of the compartment and pass through the center-line bulkhead, with a valve worked from both boiler rooms, thus forming a loop. At this forward cross connection are nozzles for 4-inch steam to dynamos, port and starboard sides; 4½-inch steam to auxiliaries forward of the firerooms, starboard side; 3-inch steam to heating system, port side; 1-inch steam to galley and pantries, port side; 2-inch steam to whistle and siren, port side.

In each boiler room the distributing castings supply steam to the auxiliary feed pumps, fireroom, fire and bilge pumps, blower engines, ash hoists, tube cleaners and connections for blowing out sea chests.

Stop valves are located on the forward side of athwartship bulkheads. The distributing castings are anchored directly to the center-line bulkhead, and the expansion in the pipes and boilers is taken care of by expansion joints, proper allowance having been made for connections on bulkheads by making bolt holes oval.

The material of pipes and flanges is the same as for main steam lines. Where condensation is apt to accumulate, drains are fitted and piped to automatic traps which discharge to condensers and feed tanks.

The pipes are lagged the same as the main steam lines.

MAIN CONDENSER.

There is one main condenser in each engine room. They are 6 feet 7 inches in diameter by 15 feet 3 inches long by 13 feet between tube sheets. The shell plating is of steel, the water chests of composition; tubes of composition, 70 parts copper, 29 zinc, 1 tin; tube sheets of rolled Muntz metal, and the tube-supporting plates of sheet steel bushed with composition.

The forward water chest, being the one for the entrance and exit of the circulating water, has a division plate on the center line of the condenser. Sizes of inlet and outlet are 18½ inches in diameter, the inlet nozzle being on the bottom.

DATA FOR ONE CONDENSER.

Number of tubes	4,878
B.W.G. of tubes	16
Outside diameter of tubes, inch	0⅝
Length of tubes between tube sheets, feet and inches	13-0
Number of supporting plates	2
Cooling surface, measured on outside of tubes, square feet	10,375
Ratio of cooling surface to total heating surface	.393

DATA FROM TRIAL.

	Starbd.	*Port*
Temperature of injection water, degrees Fahrenheit	44.0	44.4
discharge water, degrees Fahrenheit	101.0	101.6
Ratio of cooling surface to I.H.P. of main engines	1.05	

AUXILIARY CONDENSER.

There is one Blake horizontal combined air and circulating pump with surface condenser in each engine room.

DATA.

Diameter of steam cylinder, inches	6
air cylinder, inches	10
water cylinder, inches	10
Length of stroke, inches	12
Diameter of circulating-pump suction, inches	5½
discharge, inches	5½
air-pump suction, inches	4
discharge, inches	3
Cooling surface of condenser, square feet	550
Number of tubes	395
Outside diameter of tubes, inch	0⅝
Length of tubes, inches	104
B.W.G. of tubes	16

MAIN AIR PUMPS.

The main air pumps are of twin, vertical-beam type, of Blake design. They are located on the outboard side of the engine rooms, between the main condensers and the feed and

filter tank. The discharge is into the filter portion of the feed tank, and there is a 5½-inch cross connection between the suction pipes from the main condensers.

DATA FOR ONE AIR PUMP.

Diameter of steam cylinders, each, inches	12
water cylinders, each, inches	30
Stroke, inches	18
Diameter of steam piston rods, inches	2½
air piston rods, inches	2½
suction nozzle, inches	11
discharge nozzle, inches	9½

DATA FROM OFFICIAL TRIAL.

	Starbd.	*Port.*
Average strokes per minute	19.0	18.0
vacuum, inches	28.0	27.9
I.H.P.	13.7	13.3

MAIN FEED PUMPS AND FEED SYSTEM.

There are two Blake vertical, plunger, double-acting, single, center-packed main-feed pumps in each engine room, located on the center-line bulkhead forward of the operating platform. The suction is from the feed-tank connecting pipe, and the discharge from the two pumps after uniting in one 5½-inch pipe leads forward, passing through a grease extractor, which may be by-passed, and then through the feed-water heater, which also may be by-passed, into the after fireroom, reducing in size as each fireroom is passed.

DATA FOR ONE PUMP.

Diameter of steam cylinder, inches	13
water plunger, inches	9½
Stroke, inches	12
Diameter of piston and plunger rods, inches	2½
valve, inches	4½
suction nozzle, inches	5½
discharge nozzle, inches	5½
Material of water end	Composition.

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average double strokes per minute	26.5	31.5
I.H.P. per minute	59.0	61.0
pressure on feed main, pounds	325.0	325.0

FEED-WATER HEATER.

There is one horizontal feed-water heater on the discharge side of the main feed pumps, on the center-line bulkhead at the forward end of each engine room. The material of water heads is cast steel, and that of the tubes, tube sheets and shell, wrought steel.

DATA FOR ONE HEATER.

Total heating surface, measured on the outside of the tubes, square feet	830
Total number of tubes	756
Outside diameter of tubes, inch	00⅝
Length of tubes between tube sheets, feet and inches	6–09
B.W.G. of tubes	16
Thickness of tube sheets, inch	00¾

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Temperature of feed water, degrees Fahrenheit	165.0	164.4
Pressure on feed line, pounds	325.0	325.0
Exhaust steam pressure in heaters, pounds	7.5	6.25

FEED AND FILTER TANKS.

The feed and filter tank is located on the outboard side of the after end of each engine room. It is rectangular in shape, and at the top is arranged as a filter, the filtering material being "loofa," which is placed between perforated plates. The filtering chamber is divided by vertical division plates, arranged so that the water flows under and over in succession. All material is steel, well galvanized before assembling.

Capacity of each feed and filter tank, gallons	5,475
Total capacity of feed and filter tanks, gallons	10,950

AUXILIARY FEED PUMPS.

In each starboard fireroom, located on the center line bulkhead, is one Blake, vertical-plunger, double-acting, single, center-packed auxiliary feed pump. Size of pumps, 11 inches by 8 inches, and 12-inch stroke.

The suctions are from the bottom blows, athwartship compartments, C. & R. pipes from ship's sides (after pump only), feed-suction pipe, reserve-feed water tanks, hose connection.

The discharges are to the auxiliary feed main, main and reserve-feed tanks (after pump only), hose connection, overboard, in port compartment, fire extinguisher.

FIREROOM FIRE AND BILGE PUMPS.

In each port fireroom, located on the center-line bulkhead, is a Blake, vertical-piston, double-acting, single, fireroom fire and bilge pump. Size of pumps, 11 inches by 8 inches by 12 inches.

The suctions are from the bilges, adjacent athwartship compartment, sea, C. & R. drainage manifolds, hose connection. The discharges are to the fire main, overboard in same compartment, hose connection, and fire extinguishers.

MAIN CIRCULATING PUMP AND ENGINE.

The main circulating pump and engine is located on the outboard side of the forward end of the engine room. The pump has three suctions, viz: the engine-room bilge, 18¼ inches; main drain, 15½ inches; and main injection, 18½ inches. The valves of the various systems are locked in such a manner that only one may be operated at a time.

DATA FOR ONE PUMP AND ENGINE.

Capacity of pump, gallons per minute	12,000	
Diameter of suction nozzle, inches	18¼	
discharge nozzle, inches	18¼	
impeller, inches	56½	
Width of tip of blade, inches	5¼	
Diameter of H.P. cylinder, inches	10	
L.P. cylinder, inches	20	
Stroke, inches	10	
Diameter of piston rods, inches	2¼	
Length of connecting rods, center to center, inches	25	
Valve travel, inches	3	
Angular advance, H.P., degrees	33	
L.P., degrees	34	
	Top.	*Bottom.*
Steam lead, H.P., inch	1/16	1/4
L.P., inch	3/32	1/16
Cut-off, H.P., per cent. of stroke	0.77	0.715
L.P., per cent. of stroke	0.78	0.71

Diameter of H.P. piston valve, inches	4½
Size of L.P. slide valve, inches	1¼×19½
Diameter of crosshead pin, inches	3½
Length of crosshead pin, inches	5
Diameter of crank pin, inches	4½
Length of crank pin, inches	6¾
Diameter of crank shaft, inches	4½
Length of bearings, inches	5¾×8 and 8
Crank angle, degrees	90
Area of crosshead slipper, square inches	60

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average revolutions per minute	140	133
I.H.P.	52.3	44.5

EVAPORATING AND DISTILLING PLANT.

The evaporator room is located amidships on the berth deck, forward of the engine-room hatches, the distillers being in the ventilator trunk to this room on a level with the main deck.

The plant consists of three evaporators and two distillers, having a designed capacity of 16,500 gallons of potable water in twenty-four hours. The distiller circulating pump is located in the forward end of the starboard engine room, fresh-water and evaporator-feed pumps in the evaporator room.

The evaporator shells and heads are of steel plate, the front steam heads for the coils being of cast steel and the back heads of composition. The tubes are brass, rolled into Muntz-metal tube sheets. Each evaporator is provided with steam gauges and relief valves for the shell and coils, also water column, gauge cocks and blow-off valves.

The distillers are vertical and of the Bureau type. The shell is of cast iron, the covers and tube sheets of composition, and tubes of brass, tinned inside and out. The tubes are expanded into tube sheets by rolling, and expansion is taken care of by having the bottom tube sheet made with a flange that works in a stuffing box. The circulating water enters at the bottom, passes through the tubes and is discharged at the top; the vapor enters the shell at a point close to the top

tube sheet, the water leaving at a similar point near the bottom tube sheet. Baffle plates are provided for the inlet nozzles for the circulating water and vapor.

The following pumps of the Blake type are provided: Distiller circulating, size, 12 by 14 by 12 inches; distiller fresh-water, size, 4½ by 5 by 6 inches; evaporator feed, size, 4½ by 5 by 6 inches.

Piping.—Steam for the evaporators and pumps is taken from a reducing valve located in the evaporator room at the terminus of a branch from the auxiliary steam line. The vapor may be sent to the distillers or the auxiliary exhaust line, the evaporators being arranged for operation in single effect. The fresh water from the distillers passes through a meter into a small reservoir tank located in the evaporator room, from which it may drain by gravity to the main and reserve-feed tanks or may be pumped into these tanks or the fresh-water tanks. The distiller circulating pump has an independent connection to the sea, and, in addition to discharging to the distillers, may discharge to the sanitary system or fire main. The evaporator-feed supply is taken from the suction side of the distiller circulating pump.

A test was made of the plant when the ship was lying at anchor in the harbor at Delaware Capes, December 6, 1906. First each evaporator and distiller was given a two-hour trial, and then the entire plant was run for two hours, thereby furnishing an equivalent full-capacity test of four hours. The discharge from the distillers was run into the reservoir tank and a careful record kept of the number of full tanks. The water was tested for purity at frequent intervals with a solution of nitrate of silver, the concentration of water in the evaporators every half hour, and all other conditions recorded every fifteen minutes.

The following are the averages of the data obtained from the two-hour test of entire plant:

Steam pressure in evaporator coils, pounds	58
shells, pounds	12.5
at reducing valve, pounds	122
Pump strokes per minute for entire plant :	
Evaporator feed pump	24
Distiller circulating pump	32
Output of distilled water, gallons	1,619
Capacity of distilled water in twenty-four hours, gallons	19,428

DATA FOR ONE EVAPORATOR.

Heating surface, tubes only, square feet	277
Gallons of water evaporated per square foot of heating surface in twenty-four hours	20.97

DATA FOR ONE DISTILLER.

Cooling surface, tubes only, square feet	165
Gallons of water distilled per square foot of cooling surface in twenty-four hours	58.84
Gallons of water distilled in twenty-four hours	9,714
Temperature of sea water, degrees Fahrenheit	42
fresh-water discharge from distillers, degrees Fahrenheit	94
external air, degrees Fahrenheit	44
evaporator room, degrees Fahrenheit	100
Concentration of water in evaporators by hydrometer test, degrees Fahrenheit	$0\frac{2}{32}$
Quality of water distilled	Good.
Height of water in evaporators above bottom of glass, inch	1

REFRIGERATING PLANT.

The refrigerating plant is located on the starboard side of the berth deck forward of the forward turret, the refrigerating rooms, six in number, being aft of it, and the scuttle butts on the gun deck. The rooms are isolated by air locks and are insulated with cork covered on the inside with zinc, except the floors, which are covered with sheet lead.

There are two ice machines of the Allen dense-air type, each designed for a cooling effect equivalent to two tons of ice in twenty-four hours. The piping for the cooling air is arranged so that it may be sent through the refrigerating rooms and returned to the machine, or through the scuttle-butts, returning to the machine, or through the ice-making tanks before going to the refrigerating rooms.

During the test of the plant the air was sent through the ice-making tanks, then to the refrigerating rooms and scuttle butts in succession.

DATA FROM TRIAL.

Duration of trial, hours	8
Temperature of water to be frozen, degrees Fahrenheit	53
circulating water, degrees Fahrenheit	39
atmosphere, degrees Fahrenheit	41
room, degrees Fahrenheit	73
water in scuttle butts, degrees Fahrenheit	62
drawn from scuttle butts (at end of three hours), degrees Fahrenheit	54
Quantity of ice in twenty-four hours, pounds	950
water cooled in twenty-four hours, gallons	2,376
Initial temperature of refrigerating rooms, degrees Fahrenheit	60
Final temperature of admiral's room, degrees Fahrenheit	21
captain's room, degrees Fahrenheit	22
ward-officers' room, degrees Fahrenheit	8
junior officers' room, degrees Fahrenheit	17
warrant officers' room, degrees Fahrenheit	22
crew's rooms, degrees Fahrenheit	23
Total wall area of rooms, square feet	1,161
cooling surface of coils, square feet	521
Steam pressure, pounds	52
Compressor-cylinder pressure, pounds	230
Expander-cylinder pressure, pounds	52½
Revolutions per minute	118

MACHINE DATA.

Diameter of steam cylinder, inches	9½
compressor cylinder, inches	7
expander cylinder, inches	5½
piston rods, inches	1⅛
Length of connecting rod, center to center, inches	26½
Stroke, inches	13

Electric Plant.—The electric plant consists of eight 100-kilowatt generating sets (125-volt pressure at the terminals), direct connected to engines of the vertical, cross-compound, double-acting, enclosed type. Generators and engines are of the General Electric Company manufacture.

Four of these sets are located in each of the two independent dynamo rooms, one forward and one aft, on the upper-platform level.

In each dynamo room there is a main dynamo-switch board which controls the generators in their respective rooms.

Steam is taken from the auxiliary steam line, and each dynamo plant has a combined air and circulating pump and condenser for the exclusive use of the dynamo—that for the forward plant being located in a pocket forward of the boiler compartments, and that for the after plant in the passage on the starboard side between the engine and firerooms. There is a steam separator in each room, which drains to a trap and the steam end of the air pump. The exhaust is into either the dynamo condenser, main or auxiliary condenser or the atmosphere.

Adjacent to each dynamo room there is a distribution room in which is installed a distribution-switch board which is energized from either of the two main dynamo-switch boards. From the distribution-switch boards all the feeders which supply energy to the lighting and power systems are led.

The lighting system comprises a total allotment of 1,100 electric fixtures, not including special outlets for signal lanterns, truck lights, Ardois signal sets and outlets for special fixtures. There are two arc lamps in each of the engine rooms and firerooms.

There are six 30-inch searchlights, two being installed on the forward bridge, two on the after bridge and one on the searchlight platform on each of the masts. Separate and distinct from the lighting system is the power system, which supplies energy to all auxiliary machinery run by electric motors.

Included in the power system are the following: Fourteen 3-inch ammunition-hoist motors; twelve 7-inch ammunition-hoist motors; four ammunition conveyers; thirty-three ventilator fans; four pumps; six deck winches; two boat cranes; laundry machinery; workshop machinery; two 12-inch turret-turning equipments; four 8-inch turret-turning equipments; two 12-inch turret-power equipments; four 8-inch turret-power equipments; forty-one power doors, and five power hatches.

A complete system of wireless telegraphy is installed. Besides the lighting and power system there is an interior-communication system, which includes call bells, general alarms, shrill whistles, fire alarms, voice tubes, telegraphs and indicators.

A system of mechanical telegraph is also installed for communication between the bridge and engine room.

DATA.

Type of engines	Vertical cross-compound.
Number of revolutions per minute	350
Steam pressure, pounds	150
Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	18
Length of stroke, inches	10
Diameter of piston rods, inches	2½
Number of generators	8
Capacity of each generator, in kilowatts	100
Dynamos	10 pole, 125 volts.

DATA FOR ONE DYNAMO CONDENSER.

Make	Blake.
Type, Horizontal, combined air and circulating pump, with surface condenser.	
Diameter of steam cylinder, inches	10
air cylinder, inches	14
water cylinder, inches	16
Length of stroke, inches	12
Cooling surface of condenser, square feet	970
Number of tubes	800
Outside diameter of tubes, inch	00⅝
Length of tubes, inches	91
B.W.G. of tubes	16

ASH HOISTS.

The ash-hoist engines were designed and built by the contractors. They are located entirely within the ventilator trunks, and are designed to stow in one corner of the trunk within a plane not exceeding twelve by eighteen inches. The hoist will deliver to and may be operated from either the main or upper decks, the bucket-guide rails extending from the bottom of the ventilator trunk to a point well above the upper deck, with trolleys at the main deck for delivering ashes to

the chutes at the ship's side. On the test 300 pounds were hoisted from the fireroom floor to the upper deck in five seconds, including starting and stopping, with 100 pounds steam pressure.

The principle of the engine is a long cylinder taking steam at both ends through a piston valve that is controlled by the operator through the medium of a spiral spindle worked by a wheel and gearing outside the ventilator trunk. The spiral spindle works in a nut carried by the crosshead, thus putting the valve automatically on the center and stopping the engine when the operator stops turning. At the proper height above the steam cylinder is the top sheave bracket, between which, on guide rods, the crosshead travels. This crosshead is fastened to the piston rod and carries multiple sheaves corresponding to those carried by the top bracket, that give a maximum bucket travel of 50 feet for a piston travel of 6 feet 9 inches.

DATA FOR ONE ENGINE.

Diameter of cylinder, inches	7
Stroke, inches	8½
Diameter of piston valve, inches	1¼
Travel of piston valve, inches	2
Diameter of piston rods, inches	1⅜
rope sheaves, inches	8
Pitch of spiral spindle, inches	13
Number of rope sheaves	3
ash-hoist engines in ship	6

FORCED-DRAFT BLOWERS.

The forced-draft blowers are located in the firerooms, being suspended from the protective deck about over the center of the stokehold. The engines and fans were designed and built by W. D. Forbes & Co.

The arrangement is the same for all of the firerooms, there being two engines, each driving a fan located athwartship, so that the air is discharged directly to the stokehold floor at a distance of about two feet from the front of the boilers. The supply is taken from the fireroom ventilator ducts, which are closed at the bottom when under forced draft.

DATA FOR ONE BLOWER ENGINE.

Number of cylinders	2
Diameter of H.P. cylinder, inches	5
L.P. cylinder, inches	10
Stroke, inches	6
Diameter of H.P. valve, inches	2
L.P. valve, inches	4
Valve travel, inches	1¼
Diameter of piston rods, inches	1⅜
valve stems, inch	0¾
crank shaft, inches	3
pins, inches	3
fan shaft, inches	2¾
Length of connecting rod between centers, inches	15
Number of bearings	4
Total length of bearings, inches	22½
Crank angle, degrees	180
Base area of engines assembled, inches	30×30
Height of engines assembled	54½

DATA FOR ONE BLOWER FAN.

Diameter of fan, inches	66
Number of blades	6
Width of tip of blades, inches	15
Area of induction nozzle, square inches	1,017
eduction nozzle, square inches	1,385

DATA FROM OFFICIAL TRIAL.

Average revolutions per minute	314
air pressure in firerooms	.596
I.H.P. for one blower	8.13
twelve blowers	97.6

HEATING SYSTEM.

The heating system is divided into two sections, one forward and one aft, the forward section being divided into five radiator circuits and one galley and pantry circuit, the after section being divided into six radiator circuits and one galley and pantry circuit. Steam for radiators is supplied from the auxiliary steam line, the connection at the line being fitted with pressure gauges, stop and relief valves and reducing valves, set at 30 pounds. From this point the steam is carried to distributing manifolds and divided into circuits, as

mentioned above, each circuit being arranged with a stop valve so that it can be operated independently of the others.

The galley and the pantry circuits also receive steam from the auxiliary steam line, and are fitted with valves and gauges as on radiator lines, but are entirely independent, being constantly under steam.

The circuits supply steam to laundry, bakery, galleys, pantries, water heaters in bath and wash rooms, and also a coil located in fresh-water tank to prevent water from freezing.

The drains from radiators are also divided into circuits and lead to four receiving manifolds, two being located forward and two aft, each circuit being arranged with a check and stop valve to prevent water from one circuit backing up into another. The discharges from manifolds lead to traps in engine rooms, each manifold having an independent trap, and the drains from these traps are connected together and lead to a drain manifold. This manifold has two outlets, one leading directly to filter tank and the other to a manifold which discharges to main condenser, auxiliary condenser or filter tank at will.

The drains from galleys and pantries lead directly to traps in engine rooms, each section being provided with an independent trap, and these traps discharge to drain manifold mentioned above. The drains from water heaters in bath rooms have no connection whatever with other drains, and are carried directly to the auxiliary-exhaust lines.

The circuit steam and drain pipes consist of seamless-drawn brass pipe, iron-pipe size, all pipes up to 1½ inches being connected together by composition fittings, but above that size composition flanges are used. Connection at watertight bulkheads are made with composition stuffing boxes, and copper U bends are installed throughout the lines to provide for expansion.

Particular care has been taken to prevent water hammer, and as all drains lead downward no pockets have been formed.

The radiators consist of coils made up of 1-inch seamless-drawn brass pipe, iron-pipe size, and composition fittings, stop and air valves.

The stop valves have stems of triangular cross section, valve-stem guards, socket keys for opening and closing, and unions on outlet end of valves so that radiators can be taken down without disturbing the lines.

Large radiators are divided into sections, each section having an independent steam and drain valve, so that in mild weather it is not necessary to have steam on the entire radiator.

MACHINE SHOP.

The machine shop is located amidships on the berth deck, forward of the after 12-inch barbette and between the machinery hatches, access to it being from the athwartship passage, and from the engine rooms through their respective entrance hatches. The machines are arranged with line and counter shafting driven by a 15-horsepower General Electric motor. On a test made to determine whether the motor was sufficiently powerful to do the work, with all machines working on hard cast iron at a heavy cut, 6.21 horsepower was developed.

The following machines are installed:

1 28-inch extension gap lathe;
1 14-inch engine lathe;
1 16-inch crank shaper;
1 31-inch radial drill press;
1 16-inch sensitive drill press;
1 30-inch grindstone;
1 universal milling machine;
1 12-inch emery grinder;
1 combined hand punch and shears;
6 bench vises.

The tools are provided with the most modern attachments, including scroll and drill chucks, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

NOTES.

THE SIZE OF WARSHIPS.

Naval debates in the House of Lords are most academical in character, but they are often not less informing on that account; indeed, compared with the political bickerings that are often elsewhere intruded into naval discussions, the calm examination of facts that takes place is a distinct gain. So long as the First Lord of the Admiralty is a Peer, and the voting of money rests with the Commons, the naval interest is, to a certain extent, divided, and we may look for debates in the Upper House similar in character to that which took place on Wednesday of last week. Lord Brassey, in raising the discussion, "had no fault to find with the administration." "The Opposition," he tells us, "is silent because they are satisfied with the general administration of the Navy;" and, "the supporters of the Government know that the strength of the Navy in men and ships is being adequately maintained." Moreover, "where economy has been possible votes have been cut down, and no pains have been spared to perfect the training of the Navy." What more a grateful country could ask from its rulers it is difficult to imagine; still, there are one or two things that might be inquired into with advantage; and Lord Brassey forthwith proceeded to inquire into them.

First there was the case of the *Dreadnought*. Her design had been "commended by an overwhelming weight of authority" at home, and had received still more practical recognition abroad, being followed by nearly every maritime Power—by Russia, Japan, France and the United States. Still, the *Dreadnought* cost something like two millions sterling, and such ships must be few. There were operations in war which

could be more effectively performed by many ships than by few. Admiral Sir Reginald Custance has said, "No ship, however large, can stand up against the fire of two or three battleships." It is a sweeping assertion, as it would be to say, "No small ships, however numerous, can stand up against the fire of one large battleship." Before we can reduce such maxims to their current value we must define the limits of "small" and "large." The problem, which will never be settled in peace time, at any rate, is whether two one-million-pound battleships can be made equal to one two-million-pound battleship; or whatever the suggested proportion may be.

The balance of opinion is, however, evidently against subdivision of power, for battleships have been getting bigger and bigger since the days of the *Great Harry*, and more and more costly. Concentration is still a cardinal principle in sea fighting, as on land. A *Dreadnought* engaged with two one-million-pound ships has but to disable one for victory to be in her hands. It is true that in the operation she may receive damage herself, but the big ship will have thicker armor and higher speed; and, moreover, there is no chance of her power being divided, as may be the case with the two ships, which may lose the advantage of concentration. Sir William White, who has not in practice been averse to big battleships, if one may judge by his creations, has suggested that the armament of the *Dreadnoughts* might be distributed between three vessels equal in speed, armament and defence, and possibly superior in fighting power to the larger ships. The question arises, Can the three hulls and machinery, equal in efficiency to two *Dreadnoughts*, be built for the same expenditure? The guns and their mountings involve a large part of the cost of the *Dreadnought*—about 30 per cent. of the total; and it is well known that the cost of the hull of a ship does not decrease *pro rata* with size, nor does the power and cost of propelling machinery decrease with the displacement weight, the speed being constant.

No one, we think, would be more ready than the great

naval constructor to acknowledge that considerable sacrifice must be made by replacing two units by three, apart altogether from the financial question. Sir William apparently holds that the sacrifice would be warranted; and it might, of course, take one of many directions: the smaller ships would need greater power for equal speed, and the weight of hull structure would be greater for the same armament, whilst thicker armor would be more easily provided for the bigger ships. If we take cost as a basis of comparison, the advantage in these respects lies with the two vessels as against the three; and in many other things, such as up-keep, coal endurance, etc., the big ships are to be preferred. In regard to maneuvering in battle, there is more room for question, and it is a question the naval tactician must decide.

On the other hand, two fortunate shots from mines or torpedoes would annihilate the whole force if it consisted of two larger ships, whilst in like case of three smaller ones, there would be a ship left; but against this, three ships are more likely than two are to hit a floating mine. Much the same thing may be said of strandings or other accidents of navigation. In strategy the question of size is not easy to decide. Two ships can only be in two places at once, whilst it might be necessary to occupy three, or it might be desirable to proceed to a rendezvous by three routes. As battleships are, however, primarily intended for the line-of-battle, this is not so important a consideration as with cruisers. The big ship will, however, have higher speed and wider radius of action than any number of smaller ones, other features of design being proportionate.

For cruisers the problem of design is even more complicated than it is for battleships, as the duties of the former are more varied. We have gone on from the days of the *Blake* increasing the size and power of cruisers, until many of them have become really modified battleships; but naturally the cost has gone up also. A cruiser that costs about a million and a half is evidently too expensive a vessel to carry out the duties of the old frigates. Lord Tweedmouth, repeating Lord

Brassey, says the protected cruiser is almost a non-combatant; but the seme might be said of all vessels in regard to others appreciably stronger than themselves. The old sailing frigates, which were never classed as non-combatants, were not supposed to stand up to battleships, although on one occasion a frigate did fire on a battleship, and was promptly sunk for her impertinence. Lord Brassey is satisfied that the British Navy has an overwhelming superiority in protected cruisers; as, indeed, it should have, considering the overwhelming superiority of the maritime commerce it would have to protect. Lord Tweedmouth, who naturally supports the big battleship, says, in quoting naval opinion, that the big armored cruiser is more likely to be of advantage in war than the smaller cruiser; and he is of opinion that no foreign cruiser would be able to stand up against the *Invincible*. One may readily agree with this, but it does not follow, as appears to be thought by some, that we should not have cruisers capable of standing up against weaker vessels than *Invincibles*. Naturally, we would like all our cruisers to be *Invincibles*, if we could afford it. In some respects the cruiser problem is the reverse of the battleship problem, inasmuch as dispersion is a virtue with this class of vessel.

Lord Brassey has no good word to say for the eight scouts that have lately been built. They are, he says, costly but not satisfactory; flimsy in construction, and weak in armor and armament, speed being their only merit. He also disapproves of the special torpedo-boat destroyer *Swift*, which is being built by Messrs. Cammell, Laird & Co., the estimated cost of which is over a quarter of a million. This vessel is to be 345 feet long by 34 feet 2 inches beam, and is estimated to have a displacement of 1,800 tons. With 30,000 horsepower she is designed to steam at 36 knots, and will carry 180 tons of oil fuel. Lord Brassey evidently considers that the usefulness of such a vessel will not be equal to her cost, and would prefer two ocean destroyers, which could be built for about the same money. It certainly cannot be claimed for craft costing a quarter of a million sterling that numbers can be cheaply

expended in time of war. The advance in torpedo craft is, however, only characteristic of that in all vessels of the fleet, from pinnaces to battleships. In default of a design for an effective cruiser for scouting duties, which Lord Brassey says "the ablest constructors, both in the Navy and outside," have failed to produce, he would call attention to the subvention of mail steamers. Lord Brassey has always taken an interest in naval auxiliaries ever since he raised the first naval volunteer corps, about thirty-five years ago, and the principle of merchant-steamer subvention owes much to his efforts. He quoted Admiral Lord Charles Beresford's evidence before the Committee on Subsidies in support of his views.

Auxiliaries are, however, not generally considered with favor by the Sea Lords. The Royal Naval Artillery Volunteers, an enthusiastic and hard-working body, were disbanded with scant courtesy some years ago; and the present body cannot be said to have received enthusiastic support. Lord Tweedmouth now says that mercantile cruisers "can be relied on very doubtfully," as "they must be extremely fast, or they will not be able to escape from ships of more formidable type." It would be instructive to compare the maintained sea speed and coal endurance of the protected cruisers—which are, the First Lord says, "almost non-combatant ships"—with the class of ocean liners that would be available for scouting duty and conveying information in time of war. No doubt wireless telegraphy has completely modified the practice of scouting in the present day, but speed and coal endurance are still essentials to effective service of this nature. We have yet to learn how far "wireless" will stand the rough test of war. The First Lord, however, acknowledges that "if any mercantile cruisers are to be capable of being of service in war," they would be vessels like the two new Cunarders.

The most important feature of the debate was, however, the explicit statement by the First Lord of the Admiralty as to the policy of his Government. Fears have been expressed—they may have been founded on rumors originally circulated

for political purposes—that the present Administration was careless of the position the country has held as a naval power. The rumors at first gained some color from the reductions made in the Navy Estimates. It must be remembered, however, that these reductions did not originate with the present Administration, and there is nothing whatever to show that they are due to political subserviency on the part of the Board of Admiralty. Although a clear political horizon may never be taken as a reason for slackness in preparation for national defence, it is but reasonable to put forth additional effort if complications abroad threaten. A few years ago—the period of our "splendid isolation"—such complications did threaten, and the Navy Estimates went up by leaps and bounds. Since then the alliance against us, which was mostly to be feared, has ceased to threaten. The purchase of two powerful warships on the eve of the Russo-Japanese war swelled the expenditure of the year by a large amount, and might be taken as a reasonable cause for some reduction to follow.

Making all allowances, however, a reduction of the Fleet below the strength needed to maintain our naval supremacy would be a dangerous experiment that the country would justly look on with alarm; but this Lord Tweedmouth in the Lords, and Mr. Robertson on the same day in the Commons, assure us is not to take place. The First Lord of the Admiralty said that the axiom must be accepted by everyone "that our country must at all events keep the command of the sea, without having any regard to the cost at which that may be done." Mr. Robertson, speaking somewhat later on the same day in the House of Commons, said: "As to the two-Power standard, I would say—and I believe it has already been stated elsewhere by the First Lord of the Admiralty—that it is the determination of the Government to maintain our naval supremacy as it stands now, and in the event of the Hague Conference proving abortive, to hold ourselves ready to take steps to maintain our relative position."—"Engineering," April 26, 1907.

LIQUID FUEL FOR STEAMERS.

The steamer *Elax*, one of the well-known fleet of oil-carrying vessels owned by the Shell Transport and Trading Co., Ltd., which has undergone extensive repairs and overhaul by Messrs. Smith's Dock Co., Ltd., has had an official trial and demonstration of the efficiency of the liquid-fuel-burning installation fitted by the latter firm. The trial proved eminently satisfactory to all concerned, steam being easily maintained at the required pressure to propel the vessel at a speed of 11 knots on a consumption of 14½ cwts. of liquid fuel per hour. This is equivalent to a consumption of only 17½ tons per day, for all purposes, as against 30 tons of coal per day consumed for the same work on her previous voyages, a fact of considerable interest to shipowners and engineers. The *Elax*, one of the medium-sized vessels of the Company, is 4,015 tons gross register, and capable of carrying some 5,500 tons of cargo, and is the eighteenth steamer of this kind fitted for burning liquid fuel. The installation fitted by Smith's Dock Company includes the Rusden-Eeles patent burners, and the Flannery-Boyd patent system of heating coils and gravitation tanks for purifying the fuel before being conducted to the burners.—"Page's Weekly."

A LESSON IN MARINE-ENGINE DETAIL.

One sometimes wonders, when one hears of serious results springing from small causes, if we shall ever get through the elemental stage of marine-engine practice. Sea-going engineers would do much towards reaching this desired end if they would oftener take steps to put on public record the mishaps which occur to machinery under their charge, and the way in which accidents are remedied. An excellent example of this nature is given in the current number of that useful service publication the "Royal Naval Artificer Engineers' and Engine-Room Artificers' Review." The detail that gave trouble in two of His Majesty's ships was the keep-ring; the particulars given are as follow: The *Argonaut*, first-class

cruiser, homeward bound from China, reached Malta last September, and on leaving that port proceeded to carry out a passage trial. The trial commenced with 125 revolutions, 18 knots being logged. After the run had continued for some time there was a loud report in the starboard engine room. The engineer artificer at once gave the order "Center the links," and the next moment the engine room was full of steam; whilst the senior engineer rushed on deck and closed the master-valve to that set of engines. All men were cleared out and the watertight doors were closed. The port engines continued revolving at 130 revolutions. It was found, ultimately, that the mishap had occurred in the forward low-pressure cylinder. Nothing could be seen externally, but on moving the engines very slowly a clicking noise was heard with each stroke of the engine. The steam which filled the engine room was accounted for by the high-pressure receiver valve lifting when the links were centered. It was thought at first that the spring-ring was fractured, but a card taken was normal. The starboard engine was kept going until nearing Gibraltar, when the cylinder was opened up, and it was found that half the keep-ring had become displaced, having been carried into the port, where it was sheared off, thus accounting for the report.

The second mishap of this nature reported in the same publication occurred a short time previously to a smaller vessel. The third-class cruiser *Flora* was cruising at 48 revolutions when the starboard high-pressure engine gave a succession of heavy knocks. The engines were eased, and the trouble had apparently disappeared, but on increasing to above 70 revolutions, it was renewed. As the ship was in company with the *Cambrian*, another third-class cruiser, the port engines were increased to 95, the starboard engines not being allowed to exceed 70, revolutions. On opening up, it was found that the high-pressure piston keep-ring had quite disappeared. Further search revealed parts of it, with one stud, in the steam port, other parts were jammed on the exhaust side of the slide-valve and the strengthening vanes of the

valve, whilst some pieces were found in the eduction pipe and the exhaust side of the medium-pressure slide valve. There were in all 33 pieces, varying from 1 inch to 10 inches long. The ring was 1⅜ inches by ⅜ inch. The actual damage done was a slight roughening of the exhaust edge of the high-pressure slide valve, whilst the medium-pressure slide valve had a large piece of the baffle broken off. This was found in a pocket just below the mouth of the medium-pressure eduction pipe.

A keep-ring is not, of course, a part actually necessary to the working of an engine, it being fitted as a matter of precaution to prevent accident. When, however it is not securely fastened its acts in a distinctly opposite manner, as these two accidents show. One cannot but admire the courage of the chief engineers of both these ships in not advising their respective captains to have the engines stopped—they are both twin-screw vessels—for an immediate examination. No one, in either case, knew what was the matter, though something of a serious nature had undoubtedly occurred; and this might, at any time, have resulted in no inconsiderable disaster. We must, however, confess that had such an incident occurred to a vessel that was our property, we should have preferred a more cautious procedure, and our leaning that way would have been much strengthened had we been on duty in the engine room. Even if it had been known that it was only the keep-ring, it would not have been much more reassuring, for certainly one could hardly expect that substantial pieces of metal would go through two engines in the manner described without doing very much more damage. However, officers and men of His Majesty's ships are expected to run more risks than mere civilians; and we know it is a point of honor in the engine room—and a very salutary one—not to ask to stop the engine excepting under dire necessity. It would seem that there is need for a more stringent general order that all mishaps of a serious or threatening nature should be inquired into immediately, unless there should be present some extremely urgent reason to the contrary.

In the case of the *Argonaut* a new ring was made, and the voyage was continued across the Bay of Biscay. In the original ring square studs had been fitted in the piston, corresponding holes being made in the keep-ring, the latter being secured by $\frac{3}{16}$-inch split pins. One would have thought, considering the light, though important, duty the ring is called upon to perform, this would have been security enough; and this was evidently the opinion of the Admiralty and the contractors—the Fairfield Company. Events, however, showed this not to have been the case, and probably the split pins were sheared through by constant repetition of stress due to the inertia of the ring. In the case of the *Flora*, the engines for which vessel were made at Barrow, the ring was secured to the junk ring by five $\frac{5}{8}$-inch square-headed studs, and the former was held in position by split pins. When a new ring was made the edge was reduced to $\frac{1}{4}$ inch in thickness, and this gave space to allow nuts to be screwed on the studs and secured by split pins. What precaution was taken to secure the new ring of the first-class cruiser's engine is not apparent from the narrative; but it is advised that the studs should be left longer, and should be screwed above the square to receive a nut, as well as a split pin.—"Engineering," Feb. 22, 1907.

THE MARINE STEAM TURBINE.

The various successful trials of turbine steamships made during the past twelve months have generally been hailed as further triumphs for the marine steam turbine. In many respects this conclusion is entirely justified; but, at the same time, it must be confessed that the trials left matters pretty much where they were before. The pioneer turbine boat, the *Turbinia*, established once for all that the turbine could meet and beat the reciprocating engine when applied to the propulsion of high-speed craft. Such vessels must, however, always form but a small, if important, minority of the total tonnage of the world, and we seem no nearer than we were eight years ago to the solution of the problem of adapting the turbine to

the propulsion of even the larger size of tramp steamship From the outset it has been recognized that the difficulty has lain with the propeller. Even in the case of high-speed boats the turbine has had to be run much below its most economical speed, and this has been the case although its dimensions have been enormously increased as compared with those of equal output applied to the driving of electric generators. The maximum peripheral speed of the buckets in marine work is commonly much less than half that of a turbine employed for electric lighting; and if the same general design were followed in the two cases, the turbine should be at least four times as long if equal economy were to be maintained. Moreover, in order to get this relatively low peripheral speed at a moderate number of revolutions per minute, the diameter of the turbine has to be increased, which again involves a marked augmentation in the weight of the machinery per horsepower developed. The designers have gone in this direction nearly as far as they dared, and as a consequence the total weight of the turbine machinery fitted to steamships seldom shows an advantage of more than 5 per cent. as compared with reciprocating engines. In electric-light practice the turbine unit often weighs only one-fifth to one-sixth as much as the corresponding reciprocating unit. In spite of the relatively high weight of the marine turbine, it is nevertheless, as stated, run in general much below its most economical speed; and here undoubtedly is to be found one reason for the disappointing results which have in some cases been realized in service. In no instance, we believe, has the turbine engine failed to beat, both in speed and economy, a sister reciprocating boat on the measured mile. In service, however, the latter has, in more than one instance, proved the less expensive in the matter of fuel.

As the difficulty with the turbine is to get its speed down, designers have naturally selected as their speed for calculation that aimed at on the trial trip. With turbines in which the speed is maintained constant at all loads, as in those employed for electric lighting, the hydraulic losses below the governor-valve remain a nearly constant proportion of the useful work

done, so that throttling the steam to diminish the output some 20 per cent. or so means no serious reduction in the economy. If, however, at the same time that the steam is throttled the speed of the turbine is reduced, additional losses, arising from hydraulic shock, are added to those due to throttling, and the steam used per indicated horsepower per hour may very rapidly augment, particularly if, as is invariably the case in marine practice, the turbine is much underspeeded, even at full power. It would appear, therefore, that specifications for turbine boats should lay much more stress on the economy to be attained during service conditions than on the full-speed trial, and the combination of turbine and propeller should be such as to give the best economy at the relatively low-service speed, even if this involves the sacrifice on the trial trip of the fraction of a knot or so.

An additional reason for the less favorable showing of turbine craft at sea was advanced by Professor Biles in his lectures to the Royal Scottish Society of Arts last winter. He notes that at sea the resistances corresponding to a given speed appear to be greater than in trial-trip conditions. This is not a serious matter in the case of craft fitted with reciprocating machinery, since in general with them the propeller is of ample proportions for the thrust it has to develop. With turbine craft, however, the propeller is often of such scanty proportions that quite a moderate increase in the thrust to be developed leads to cavitation and a consequent large diminution in its efficiency. The recent careful experiments made at the Navy Tank, Washington, D. C., have much simplified the problem of combining a turbine and propeller so as to obtain the maximum possible over-all efficiency. Many people are still under the impression that the percentage of slip is a direct measure of the propeller losses. In the old days this view led to the production of fancy propellers in great variety, all schemed out with the idea of reducing the slip to a vanishing quantity. But these propellers invariably proved less efficient than those they were designed to replace, and it was then pointed out by Rankine that some slip was necessary if any thrust at all were

to be developed. In spite of this, however, the false impression as to the significance of slip still survives, and promises to continue popular for many years to come.

The Washington experiments demonstrate once again that a high slip ratio, even as much as 40 per cent., is no bar to a reasonable propeller efficiency, which, even with the figure stated, may be 66 to 67 per cent. It is not improbable, indeed, that this figure might be improved on by giving the propellers an increasing pitch from the front edge to the back. It is known that at low slip ratios there is no appreciable advantage in this procedure, but theory indicates that there might be a substantial gain when the slip approaches 30 per cent. or more. One drawback to the practice would be that the efficiency in going astern would be diminished, and turbine craft are already somewhat deficient in the matter of retrogression.

Although, as stated, the Washington experiments have simplified the problem of selecting the propeller most appropriate to a given set of conditions, they have pretty conclusively demonstrated the hopelessness of any such improvement in the propeller as will permit of the ordinary reaction steam-turbine being applied to the driving direct of the propellers of the average tramp steamer. If the turbine is to come into general use for such craft, either some form of reduction gearing must be used, or some other form of turbine adopted. An analagous case is found in those water-power plants in which the quantity of water available is small, whilst the head is high. In such cases reaction turbines are never used, as the requisite speed of rotation becomes enormous, but some form or other of impulse turbine working with partial admission. With water such turbines have very high efficiencies; but partial admission in the case of a steam turbine involves some serious losses, since the wheel has to run downward. Nevertheless, it is perhaps on these lines that the application of the turbine to small-powered craft will ultimately be developed.

At first sight the system of partial-admission impulse tur-

bines, velocity compounded on the Ferranti or Curtis system, would appear to possess great advantages for marine work in all cases. It is, however, impossible to use as narrow blades with velocity-compounded turbines as with the pressure-compounded type. With the former, provision has to be made for a very great increase in the radial depth of the belt of steam as it flows through the blades, and if the latter are very narrow, this radial expansion, which, it must be remembered, occurs without change of pressure, would be effected so abruptly as to lead to very large eddy losses. Here, no doubt, is to be found the reason for the failure, so far, of this type to show any saving over the Parsons marine pattern in the matter of the weight and length requisite to the attainment of a reasonable economy of steam. In our issue of November 16 we reproduced a paper by Dr. Lasche, showing a marine turbine of this type designed to develop 8,000 horsepower at 400 revolutions per minute. The diameter over the tips of the low-pressure blades was 2¼ meters (7.37 feet), and the length of the working part, from admission to exhaust, about 8 feet 6 inches. The estimated steam consumption (the turbine was never built) was, we understand, about 14 pounds of steam per brake horsepower hour. This is no doubt an excellent result in the matter of compactness, combined with a reasonable efficiency; but there would be little difficulty in equalling it with an ordinary reaction turbine having properly-set blades, and the weight of the latter would be substantially less. No doubt the leakage loss at the high-pressure end of the reaction turbine would be considerable, but would not exceed that due to the extra hydraulic resistances which are inseparable from the system of velocity compounding adopted for the corresponding high-pressure end of the Allgemeine turbine. For larger powers, therefore, we think the Parsons turbine will hold its own; but there does seem a possibility that a combination of the Curtis arrangement for the high-pressure end with the Parsons for the low-pressure might possess advantages in the case of the lower powered and relatively slower vessels. A very detailed study of the

problem would, however, be necessary to definitely settle this, as in steam-turbine practice the aphorism that "general reasoning is generally wrong" acquires an exceptional validity. —"Engineering."

LIMITS OF THERMAL EFFICIENCY IN INTERNAL-COMBUSTION ENGINES.

At the ordinary meeting of The Institution of Civil Engineers, on Tuesday, the 26th February, Sir Alexander Kennedy, LL. D., F. R. S., President, in the chair, the paper read was "On the Limits of Thermal Efficiency in Internal-Combustion Engines," by Dugald Clerk, M. Inst. C. E. The following is an abstract of the paper.

The Institution Committee on the Standards of Efficiency of Internal-Combustion Engines among their recommendations as to the standard engine of comparison for internal-combustion motors, recommended that for the purpose of the standard, air—assumed to be a perfect gas having a value of $\gamma = 1.4$—should be taken as the working fluid. For the ordinary four-stroke cycle engine, the formula giving the efficiency then is

$$\eta = 1 - \left(\frac{1}{r}\right)^{0.4};$$

where $\frac{1}{r}$ is the ratio of the minimum volume to maximum volume. The Committee were satisfied that with good engines, giving their best economy, the actual efficiency divided by the ideal efficiency determined by this standard could be expressed by a ratio which varied between 0.5 and 0.7. This was deduced from separate tests made by Professor Meyer and Professor Burstall. Professor Burstall's tests also showed how inefficient design would decrease the ratio, as in some of his tests means involving greatly increased cooling surfaces were employed to increase the compression, and were found to considerably diminish the ratio. These tests showed further how too high flame temperature also decreased the ratio. The Committee required, however, further knowledge as to the

effect of the dimensions of the engine on the ratio, and accordingly they made tests on three engines of 5-inch, 9-inch, and 14-inch-diameter cylinders respectively, giving 6, 24, 60 I.H.P. In these engines, taking the mechanical efficiency to be 88 per cent., and calculating the I.H.P. from B.H.P., they found that the efficiency ratios were 0.61, .65 and 0.69 in the three engines. The tests showed, therefore, that by bearing in mind the slight changes in the ratio due to difference in dimensions, a close approximation to the best indicated efficiency to be expected from a given compression could be obtained by the use of a factor varying between 0.60 and 0.70, according to the dimensions of the engine. The tests also showed very clearly the small increase in economy of large engines in comparison with small ones, there being only 12 per cent. increase between 6 horsepower and 60 horsepower. The possible efficiency with the actual fluid used in the engine was known to be less than that given by the air standard. The Committee considered that a definitely known standard from which the actual efficiency could be deduced by using a multiplier found experimentally, allowing for the imperfections of the engine as well as for variations in the properties of the working fluid, should be adopted until the properties of the working fluid were accurately known. The author has examined the results of the test made by the Committee, and has made some further experiments on the large engine used in the tests, with a view to finding the true heat distribution in the engine.

The balance sheet given by the committee is as follows:

	L	R	X
Exhaust waste, . . .	35.3	40.0	39.5
Jacket waste, . . .	23.5	29.3	25.0
Radiation, ...	7.6	10.0	7.3
B.H.P., ...	26.7	28.3	29.8
	93.1	107.6	101.6

In obtaining this balance sheet the exhaust waste was determined by calorimeter, jacket waste measured, and the

radiation includes friction of the working parts. The B.H.P. was determined by rope brake. In order to reason as regards properties of the working fluid, it is necessary to know the I.H.P., the loss of heat during explosion and expansion, and the heat in the gases at the end of expansion. These quantities are not given in the ordinary balance sheet, as determined above. In the ordinary test the jacket loss is always over-estimated, because some heat which ought to go to the exhaust calorimeter flows to the water jacket after the opening of the exhaust valve and all through the exhaust stroke of the engine. The piston friction also will appear in the water jacket. The author has therefore attempted to adjust the balance sheet from data given in the Committee's report. Taking the mechanical efficiencies for the three engines, L, R and X, as 0.84, 0.85 and 0.86, the friction percentage of total heat is 5.1, 5 and 4.9, respectively. Deducting this from the jacket waste, corrected values for heat to water jacket, 21, 26.8 and 22.6 per cent. are obtained. Using these values, and reducing to percentage, assuming that the error in total heat is not in the I.H.P. item, a new balance sheet is obtained:

	L	R	X
Exhaust waste, . . .	41.1	37.1	39.9
Jacket waste, True radiation, . . .	27.1	29.6	25.4
I.H.P., ...	31.8	33.3	34.7
	100.0	100.0	100.0

The ideal efficiencies in these engines are practically the same, and assuming that one-third of the heat going to the engine is converted to work, and that the heat loss occurs near the beginning of the stroke, the difference between the jacket plus radiation losses in any two engines should be three times the difference between the I.H.P.'s. In the L and X engines this is found to be exactly the case. The jacket waste in the L engine is evidently too low, and on the above considerations should be 34.1. This value of jacket waste for the L engine will give an exhaust waste of 34.1, which is

practically the same as that determined by calorimeter. It appears, therefore, that in the L engine some heat which should have appeared in the water jacket has been lost. This corrected balance sheet is probably more accurate than that obtained in the test; but there is still some heat found in the water jacket which should be in the exhaust. The experiments give no means of determining this amount. The balance sheet, however, gives a method of calculating the maximum possible efficiency of the actual fluid. Adding exhaust waste to I.HP., and dividing I.H.P. by the sum, possible efficiencies for the three engines of 0.482, 0.473 and 0.465 are obtained. In obtaining these efficiency values, however, it has been assumed that the heat is lost at the beginning of the stroke, and, therefore, the values are not accurate. If the distribution of heat loss were known, the true adiabatic could be constructed and correct results obtained.

To check the results, indicator diagrams which give the correct mean pressure, have been studied. From the composition of the exhaust gases and the charge temperature, the weight of the charge is found to be 0.14 pound. From the diagram, the temperature drop from the end of expansion to charge temperature is 1,745 degrees Fahrenheit. The specific heat of the gases by weight, assumed constant, is 0.185. From these values, obtained from the numbers given in the Committee's report, it appears that 43 per cent. of all the heat of the combustible gases is accounted for in the exhaust. This gives a balance sheet for the X engine:

	Per cent.
Exhaust waste,	43.0
Jacket waste and radiation,	22.3
I.H.P.,	34.7

The exhaust waste here would obviously be greater if specific heat increases with temperature. From this balance sheet, calculated as before, an efficiency of 0.447 is obtained; with air the efficiency would be 0.49. These considerations show the difficulty in using the actual fluid as a standard. In

spite of the great labor expended on experiments, only a rough approximation to the true heat distribution can be arrived at.

In 1884 the author made experiments on cooling after explosion in a closed vessel. Many other investigators have since done similar work, but cooling of a cylinder having a moving piston had never been investigated. The author made further experiments, and determined the cooling in the X engine. The engine was run at normal speed, and when a charge had been drawn in, the rollers actuating the inlet and exhaust valves were slipped, so that the valves remained shut. The explosion then took place, and the gases instead of being discharged were alternately compressed and expanded. An indicator card gives a cooling curve, showing temperature fall during successive revolutions of the engine. From these cards the mean apparent specific heat of the gases in the cylinder has been deduced, the gases being practically the same composition as those in the Committee trials. The values given increase with the increase of temperature, and have been called apparent specific heat values, because certain facts discovered are inconsistent with the change, being entirely specific heat change. Calculations, assuming these numbers to be the true specific heats are, however, very nearly accurate. From the cooling curves and specific heat values so determined a balance sheet has been obtained for the X engine as follows:—

	Per cent.
Heat flow during explosion and expansion, . .	16.1
Heat contained in gases at end of expansion, . .	49.3
Indicated work, ...	34.6

Comparing this with that found by the Committee, it is seen that the indicated work is the same in both. There is, however, less heat flow during expansion, and more heat in the gases at exhaust. This shows that about 21 per cent. of the heat in the gases at the end of expansion goes to the water jacket during the opening of exhaust valve and exhaust stroke.

This is considered a more accurate balance sheet than has yet been obtained. Calculating the ideal efficiency as before, the value 41 per cent. is obtained. From the values of specific heat given, the adiabatic may be calculated, from which the ideal efficiency is found to be 39.5 per cent., showing that the actual engine has converted 88 per cent. of the heat which it possibly could convert into indicated work. The new method has been checked by a test of a small Stockport engine in the author's laboratory, which gave similar results to those given by the X engine.

Tables have been calculated showing the ideal efficiencies for different compressions using the specific heat values given, and showing that, roughly, the air standard is 20 per cent. too high, and that if gamma be taken 1.285 for the explosion line, and 1.37 for the compression line, the change of specific heat between 1,700 degrees Centigrade and 1,000 degrees Centigrade, commonly used in practice, is too small to produce much error. More investigation is, however, required before even the apparent specific heat values can be accurately known for the various mixtures used in internal-combustion motors.

Much has been recently done, including experiments by Professor Hopkinson, Messrs. Bairstow and Alexander and Professor Burstall; but until further knowledge is obtained the air standard as defined by the Committee gives the best basis for comparing the performances of different engines.

The appendixes show the method of calculating the suction temperature, charge temperature, exhaust temperature and charge weight, and also method of calculating the adiabatic and efficiency for varying specific heats.—"The Engineer."

THE CRUISER OF THE FUTURE.

The three new cruisers of the *Invincible* class have been launched. The first, the *Indomitable*, was floated from the Fairfield Works at Glasgow; the *Inflexible* from the Clydebank yard of Messrs. John Brown & Co., Limited; and the *Invincible* from the Elswick Works of Sir W. G. Armstrong,

Whitworth & Co., Limited. These vessels are the most remarkable cruisers yet built, alike for their gun power and speed, and may be accepted as suggestive of the Admiralty's conception of the cruiser of the future. Their design has aroused considerable interest and criticism, not alone because of their great cost, which averages 1,729,000 pounds each—equal to the price of a modern battleship—but because of a sharp difference of opinion as to naval tactics applicable to cruisers.

It is being more fully realized than formerly that even the principles of strategy must vary or progress with the change of weapons. The enormous advantage of wireless telegraphy, if not also of torpedo and submarine craft, must influence the practice of war. For instance, as Mr. Julian Corbett pointed out in his lecture last week at the Royal United Service Institution, a close blockade has now become impossible, owing to the advent of these three important auxiliaries in naval warfare. In carrying out an open blockade or investment of a port so that the watching ships shall be beyond the range of torpedo action either from surface or submarine craft, it becomes necessary to have an exceptionally high speed. Only thus may full advantage be taken of geographical conditions—*i. e.*, of temporary lairs, from whence to make a dash after the enemy in the event of his trying to escape. The size of the area within which such positions may be sought is in direct proportion to the speed. If strategy may thus be influenced by advances in connection with the improvements in *matêriel*, tactics are to be more extensively affected, and this is particularly so in connection with cruisers. Wireless telegraphy has enormously increased the possible area within a screen formed by the scouts of a squadron. As a consequence, it is impossible to ascertain the full strength of an opposing squadron within such a screen without driving home a reconnaissance in force. This necessitates powerful cruisers, so that they may come within sight of the main force of the enemy without suffering. Accurate information must be got at all costs. Unless this risk can be taken,

no admiral can be certain of the adequacy of his scouting force. The futility of depending on cruisers of deficient power, when the enemy may use as scouts the fastest of his well-gunned ships, is fully recognized. These vessels of the *Invincible* class have consequently been made the most powerful cruisers yet conceived, so far as present knowledge goes.

Indeed, these vessels more closely resemble battleships than any cruiser yet built. Their armor is only excelled by the ships of the *King Edward* class and the *Dreadnought* class. Their armament is equalled in broadside fire, or in bow and stern fire, only by the vessels of the *Dreadnought* class. Their speed of 25 knots is immensely superior to that of any armored ship afloat. As regards coal capacity, they are equal to any cruiser in the service, carrying 1,000 tons on normal draught; and although the high power necessary to give the maximum speed will make severe demands on the fuel supply, the radius of action should be satisfactory. This is the element in design which is most likely to be criticized on the ground that speed has probably been bought at some cost in respect of endurance. This raises the interesting question as to whether speed or endurance is the preferable quality in cruisers. It has been a cardinal principle in British warship design that radius of action is of greater consequence to a British fleet than to a foreign fleet, but there is suggestion now of some tendency to modification of this view. For over-sea work endurance is more essential, perhaps, than speed; but in the "narrow seas" speed is probably preferable. The number and distribution of our coaling stations greatly simplifies the problem so far as our Navy is concerned. On the other hand, the absence of such coaling stations intensifies the difficulties experienced by foreign powers. In the competitive game the more we can "force our rivals to sacrifice endurance to speed, the easier is our Imperial defense." Herein lies the strategical dilemma of the foreign naval designer, the one which has involved serious delay in the laying down of ships to meet our *Dreadnoughts* and our *Invincibles*.

The problem of the protection of merchant shipping may somewhat complicate the situation. The Admiralty view,

which must be accepted as the result of careful analysis of war experience, and therefore most reliable, is that the most effective procedure is to engage the enemy before the dissipation of his forces for the attack of commerce. Such dissipation of force, however, would be exceedingly dangerous, because the influence of wireless telegraphy must soon result in the attacking ship being mastered by a cruiser squadron, presuming even that she could keep the seas without returning for supplies to ports more or less closely invested by our fleets. There are those, however, who consider that special commerce protectors should be provided. Indeed, they contend in favor of the maintenance for the purpose of ships more or less obsolescent, and this view is based on the success of Japanese ships inferior to some of our vessels now regarded as practically obsolete. There is, as we have time and again pointed out, a danger of deducing lessons from the Russo-Japanese war which are not justified, owing to the ineptitude of the Russian ships or the special conditions prevailing. To utilize experienced officers and men in slow or defectively-armed ships, which would be at the mercy of one modern high-speed cruiser with high-velocity guns, seems to be courting disaster. The present trend is distinctly in favor of cruisers acting in squadrons in order to bring the enemy to fleet action, and to make it impossible, if not dangerous, to engage in a *guerre de course*. As Sir Charles Dilke pointed out at the London Chamber of Commerce on Wednesday evening, a neighboring naval power, after an inquiry by most competent authorities, had abandoned the view that it could, while not holding the seas against us, inflict great damage to our commerce.

The combination of qualities in the new cruisers has necessitated an exceptionally long ship. Hitherto the longest of the British modern cruisers have been the *Powerful* and *Terrible*, and the four armored ships of the *Drake* class, all of which had a length of 500 feet and a beam of 71 feet. Following the *Drake* class, completed six years ago, there was a tendency to decrease the length of the vessels, but later there has been steady advance. Thus the County class are 440 feet, the *Devonshire* class 450 feet, the *Duke of Edinburghs*

480 feet, and the *Minotaurs* 490 feet. The beam has, in the same period, been increased from 66 feet to 74½ feet. The three *Invincible* cruisers are 530 feet long and 78 feet 6 inches beam, the draught continuing, as in most of the recent armored cruisers, at 26 feet. This increase in length of 30 feet, as compared with the longest preceding cruiser, and of 40 feet as compared with the immediate predecessor, is necessary partly to enable high speed to be realized with the minimum of power, as the form for a given displacement may be finer, but it is a consequence also of the need for satisfactory disposition of the guns. It was laid down as an essential condition, first, that all the guns of the primary battery should be of 12-inch caliber—a condition never before exacted in any ship except the *Dreadnought*—and, second, that all the guns should have an arc of training to enable them to fire on either broadside and through very large arcs forward and aft. Eight guns are mounted in pairs in barbettes, one forward and one aft, and one on each broadside, but not, as in previous ships, on the same transverse line, the port barbette being some distance forward of the starboard barbette. This constitutes, as we have already indicated, a broadside armament equal to that of any ship afloat, equal even to the *Dreadnought*, and it will be interesting to show in tabular form the progress in the gun-power of successive armored cruisers.

BROADSIDE FIRE OF SUCCESSIVE ARMORED CRUISERS.

Class and year of launch.	Designed speed.	Displacement.	Number and caliber of guns firing on each broadside.	Collective muzzle energy from one round.
	knots.	*tons.*		*foot-tons.*
County (1900)	23	9,800	Nine 6-in.	30,200
Drake (1901)	23	14,100	Two 9.2-in. Eight 6-in.	63,600
Devonshire (1904)	22.25	10,850	Three 7.5-in. Three 6-in.	40,400
Duke of Edinburgh (1904)	22.75	13,550	Four 9.2-in. Five 6-in.	99,500
Minotaur (1906)	23	14,600	Four 9.2-in. Five 7.5-in.	137,000
Invincible (1907)	25	17,250	Eight 12-in.	381,576

It will be seen that in seven years the displacement tonnage has been doubled, and that the collective muzzle energy from one round of guns has increased more than twelvefold. What is of more importance, however, is that the County class are only capable of fighting at three miles' range, and even then against inferior ships. It is true that the *Drake* class of six years ago can effectively bring to bear two guns at four miles' range; the *Duke of Edinburgh* class and the *Minotaur* class, four guns of the same range; whereas the *Invincibles* will be able to utilize all of their eight guns at five miles' range, and then to do effective work against any foreign battleship. In this way they will be able to combat an equal number of battleships of the enemy's force while doing reconnaissance work, having at the same time a speed which will enable them, after gleaning all information as to the force of the enemy, to return to the admiral with full knowledge of the strength of his opposing force. Although their armor protection may not be as effective as the latest of our battleships, they will still be able to take their place in the line of battle; and on the principle that the most effective defence is an active and preponderating offence, they will do effective duty.

The value of the superior gun power is further considerably augmented, not only by the unification of the caliber of the gun for reasons which were admirably enunciated by Lieut. Comdr. Sims, but because of the exceptionally high freeboard of the ships. In the preceding cruisers—notably the *Duke of Edinburgh* class and the *Minotaur*—the forward guns were placed on the forecastle, which was cut away on each side to enable the wing guns on the upper-deck level to fire ahead. These wing guns, like all the other primary weapons in the ship, are in these earlier cruisers on the upper-deck level. In the new ships the broadside guns are on the same level as the forecastle, and therefore at a great height above the water line. The aft pair of guns alone are on the upper-deck level; but the deck erection to the rear of these stern guns is cut away at an acute angle on each side to enable the aft weapons to train as far as possible forward of the beam.

The disposition of the guns and machinery has involved difficult problems in order to secure a wide arc of training, and as this is a matter of increasing importance in the design of warships, considerable interest will be taken in the paper to be read next Wednesday at the institution of Naval Architects by Mr. James McKechnie, of Barrow-in-Furness, as his unique experience of both propelling and gun machinery entitles him to speak with authority on "the influence of machinery on the gun power of the modern warship."

As to the armor protection, this, as in the earlier ships, is continued right fore and aft, and extends from a considerable distance below the water line to the upper-deck level, the gun-machinery above this being further protected by the barbettes. The armor on the broadside amidships is, for some distance, 7 inches in thickness, reduced forward and aft to 4 inches, while all of the guns and the machinery are thus within heavy armor. Although the protection is not equal to that of the *Dreadnought*, it corresponds with the plating of the *Duncan* class, with this further advantage, that it is continued right aft as well as forward. As suggestive of the relative extent of armor protection in modern cruisers, it may be added that the proportion of weight of hull to the displacement tonnage of the ship in the County class was 60 per cent., in the *Devonshire* class 61.5 per cent., in the *Duke of Edinburgh* class 59 per cent., in the *Minotaur* 56.3 per cent., and in the *Invincible* 56 per cent. The inference may be justified that the weight devoted to protection is relatively less than in the earlier ships; but the length of the ship is a factor in such comparison, and in any case the difference is small.

The machinery is of the Parsons type, with water-tube boilers designed to give 41,000 horsepower. The disposition of the machinery has been arranged to enable a magazine for the amidship guns to be placed under the guns, and there are three boiler compartments, with two engine compartments, the latter divided by a longitudinal bulkhead. The turbines are arranged as in the *Dreadnought*. There are four shafts. On each of the center shafts there are fitted a cruising turbine

and low-pressure ahead and astern turbines, the two latter being within one casing. On each of the wing shafts there are a high-pressure ahead and high-pressure astern turbine. At cruising speed, therefore, the sequence, going ahead, will be, the cruising turbine on the inner shaft, the high-pressure turbine on the outer shaft, and the low-pressure turbine on the inner shaft; while for going astern the sequence will be, the high-pressure turbine on the outer shaft, and a low-pressure turbine on the inner shaft. There are four propellers, one on each shaft, and the outer propellers are about 20 feet ahead of the inner. The two inner shafts are carried within spectacle framing and stern bracket, and are immediately ahead of the two rudders. These, as in the *Dreadnought*, hang from the stern structure in a coned bearing, and they are further secured by nuts. These rudders are of very ample area, so that the turning power of the ship will be considerable.—"Engineering," March 15, 1907.

MARINE GAS PROPULSION.

THE DIFFICULTIES IN THE WAY OF APPLYING THE GAS ENGINE TO THE PROPULSION OF LARGE SHIPS AND A POSSIBLE SOLUTION.

By A. VENELL COSTER, Manchester Association of Engineers.

Last month we reviewed in these pages an article dealing with the possibilities of the internal-combustion engine for marine purposes which considered only the mechanism and arrangement of the engine. A paper on the same subject, by Mr. A. Vennell Coster, recently read before the Manchester Association of Engineers, contains a discussion of fuels and gas producers and suggests an arrangement of engines which differs widely from that proposed in the former article. A few extracts are taken from Mr. Coster's paper with a view to showing how he proposes to solve the difficulties in the way of applying the gas engine to the propulsion of large ships.

"I feel confident that if a marine engineer were told that by the introduction of the gas installation he would obtain the fol-

lowing advantages, he would be more than satisfied with the substitution of gas for steam power.

"(1) The ship driven with half the amount of fuel.

"(2) Stand-by losses reduced over 75 per cent.

"(3) Working pressure confined to the engine cylinders.

"(4) No boiler tubes or main steam pipes to burst, or furnace crowns to collapse.

"(5) No priming in a heavy sea way, or water hammer in pipes and cylinders.

"(6) No more difficulties with the firing of boilers in a beam sea. Gas producers may be charged only twice every twenty-four hours, and the rolling and pitching of the vessel is rather an advantage than otherwise in assisting the fuel down from the charging hoppers.

"Firms of the standing of Messrs. Beardmore, Thornycroft, Yarrow, Crossley Bros., Nobel Bros., Sulzer Bros., Mr. H. Cherry and many others have already applied themselves to this venture, and the results so far have been by no means discouraging. There are still many difficulties to be faced and overcome, but they will be overcome. In this paper my proposal is to put before you such a scheme of marine gas propulsion, that the most cautious of us would be willing to risk a voyage by its means. What are the main difficulties?

"1st. The construction of a gas producer able to gasify all grades of bituminous coal.

"2d. A simple method to cleanse the gas from tar, either before the introduction of the fuel into the producer proper; when in the producer; or after the gas has left the producer on its way to the engine.

"3d. Perfect control of the gas-propelled vessel in starting, stopping, reversing and running at all speeds.

"These are the three main difficulties in our way.

"Dealing with No. 1 and No. 2. If anthracite coal or coke is used these difficulties are entirely avoided, but with a limited supply of anthracite our wisest course is to leave it out of the question altogether. For marine purposes we must deal with the ordinary coal that may be obtained, not only in Wales, but

in various parts of the United Kingdom, and the many coal-fields scattered about the world. Bituminous coal is cheap compared with either anthracite or coke, and as I have already stated, it is the only possible fuel for marine gas producers. Its thermal value varies from 10,000 to 14,000 British thermal units per pound, and costs from six shillings to seven shillings per ton. For many years gas engines have been driven by bituminous coal gas, and the difficulty has always been to get rid of the condensible hydro-carbons which would fill and clog the valves and gas passages of any engine they were allowed to enter. This problem is one that must be faced rather by our chemists than by our engineers, and many notable firms have already solved the problem, but not in that simple manner that would make the gas plant perfectly suitable for marine purposes. It is reported from several trustworthy sources that a simple solution has been discovered.

"Then there is the necessity of dealing with the many grades of bituminous coal, some suitable, many unsuitable. Shipowners would have to insist on a correct analysis being supplied prior to delivery, so as to ensure gas producing qualities.

"The third difficulty, namely: Perfect control of a gas-propelled vessel is our next consideration.

"One of the main difficulties that confronts the gas engineer in applying gas engines for marine purposes is that of reversing. With small powers it is possible to reverse by means of a reversible propeller, the engine having only one direction of rotation. There is naturally a very serious objection to putting the reversing gear in so inaccessible a position as in the propeller, but at the same time, there is undoubtedly a future for this type for river and coast work, if made simpler and more reliable. For higher powers up to say 500 horsepower units there is another system in vogue for reversing the propeller, namely—a reversing gear, which may be a combination of bevel or spur gear and friction clutches, is fitted somewhere on the propeller shaft within the vessel and in an accessible position. But for powers above 500 horsepower the gas engines themselves must be made to reverse, for this dispenses with

expensive and complicated clutches. There is no practical difficulty in reversing the gas engine, provided a sufficient reservoir of compressed air is at command.

"But the greatest of all difficulties in the application of gas engines for marine purposes, is the attainment of a sufficient range of mean effective pressures in the engine cylinders, so that like a steam engine, varying speeds, from full speed to dead slow, can be run without fear of stopping. We may reasonably assume that a maximum reduction of sixty per cent of the mean pressure leads only to a reduction of thirty-five per cent. of the speed of rotation, or, if half the cylinders are cut out, the speed of revolution will be reduced by about thirty-two per cent. Then by reducing the mean pressure in the remaining cylinders, the result will be a reduction of the effective turning effort to twenty-five per cent. of that of full power, the speed of revolution will be something less than half that of full power. This reduction of speed is not satisfactory for marine purposes and it is doubtful whether gas engines could be run with so great a reduction of explosive charge. As an example, the marine gas engine in the *Lord Antrim* (mentioned below), with normal speed of 150 revolutions, can not safely be reduced below seventy-five revolutions per minute, because with a light fly-wheel and possible miss-fires, the engine has not sufficient momentum to carry it over the compression strokes.

"The following solution at once meets all these difficulties, whether the vessel has to run full speed or dead slow, by a practical manipulation of the speeds of independent engine units. In running some of them ahead and some of them astern any speed in either direction can be readily attained by the officer in charge of the vessel on the bridge, and without stopping, except during reversal, a single engine or the normal flow of gas through the gas producers.

"By this means the author attempts a solution (analogous to the turbine problem), but on different lines from those apparently adopted by builders of marine gas engines of 500 horsepower and over. They rely on a large reservoir of compressed air for maneuvering and working the vessel up dock, in nar-

row waters, or when running dead slow in foggy weather, because with one engine unit, as has already been stated, the vessel can not be driven by gas much below half speed. But to depend on compressed air at these times, when engines and producers should prove not their inefficiency but efficiency, is to stultify the whole subject of marine gas propulsion."

Mr. Coster illustrates his solution by means of a concrete example. He takes the *Lord Antrim,* a vessel 375 feet long by forty-seven feet beam and twenty-eight feet depth. It is at present run by one triple-compound engine, running sixty-seven revolutions per minute and developing 2,360 indicated horsepower. For this engine Mr. Coster proposes to substitute three gas engines developing the same horsepower as the steam engine, but running at the increased speed of 150 revolutions per minute, the gas being supplied by four producers with their auxiliary apparatus, these producers having sufficient capacity to supply, besides the main engines, several smaller units for the driving of dynamos, electricity being used to drive the air-compressing apparatus, the pumps and other necessary machinery.

Each of the main engines runs a separate propeller shaft. "The central shaft is fitted with a larger propeller than the two wing shafts. These have two propellers carried on **A** frames from the bilge of the vessel, and do not project beyond the lines of the vessel, and are thus satisfactorily protected. It is possible, as already intimated, with this arrangement of machinery, to run the vessel at any speed from full speed to dead slow either ahead or astern, without stopping the engines except for reversing. For example: taking the propeller going astern as having only fifty per cent. efficiency when compared with going ahead, by running the center engine full speed ahead, and the wing propellers about full speed astern, the vessel can be practically stationary. Then by slowing down the wing propellers to half speed astern, the vessel can be made to move about quarter speed ahead, and as these engines can vary their speed from 150 revolutions down to seventy-five revolutions per minute, any speed in any direction can be attained by the vessel without stopping the engines. In narrow waters, en-

tering or leaving port, or working up dock, the vessel is under as complete control as if fitted with the latest steam engines, and with no fear of the quality of gas being reduced. With such an installation, which occupies somewhat less space, less weight, and about the same cost as a steam installation, but with about fifty per cent. less coal consumption, who would fear a deep-sea voyage if the machinery were in charge of experienced and capable marine gas engineers?

"There is no question in my mind as to what type of gas engine we should adopt for marine purposes, the simplest is the best, and there is no type so simple and satisfactory as the four-cycle or 'Otto' type, with its sequence of suction, compression, power, and exhaust strokes, taking place in each cylinder. This type of engine, with inverted vertical cylinders, has a mechanical efficiency of over ninety per cent. and a thermal efficiency of thirty three per cent. and for a vessel requiring say 3,000 horse-power units of six single-acting cylinders, with pistons of say twenty-one-inch diameter by twenty-four-inch stroke, would be suitable. But where larger powers are required, then tandem cylinder engines with open cylinder liners, having piston rod and stuffing box between the tandem cylinders, where they can be easily inspected and adjusted, would be required."—"The Engineering Magazine."

HOW COMPRESSED AIR RAISED A SUNKEN SHIP.

THE REMARKABLE SALVING OF THE STEAMSHIP *BAVARIAN*.

To turn a 12,000-ton steamship into a huge steel bubble by pumping her full of compressed air is a recent engineering feat.

The steamship *Bavarian* of the Allan Line, ran on Wye Rock, thirty-eight miles below Quebec, on the night of November 3, 1905, a few minutes before high tide. Almost every method known to wreckers for salving the vessel was tried and found wanting and over $150,000 was spent in these efforts.

Examination had shown that the *Bavarian's* bottom amidships was in a very ragged condition. The holes were so large that it would be hopeless to try to pump the water out. Preparations were accordingly made to treat the holds as caissons, compressed air being used to force the water out through the opening in the bottom.

Work was begun by the North American Wrecking Company on September 7, 1906, and the vessel was floated on November 16, 1906. Everything in regard to the operations was calculated with mathematical accuracy. The calculations for the buoyancy required, and at the points chosen, were most fortunate.

It was necessary to timber solidly between decks above the several compartments that were to be used as caissons. The hatches were closed by plating. Air locks were placed on the compartments which were to be treated as caissons. Every opening in the deck, scuppers, etc., was closed.

When the air was applied the water rapidly receded and workmen were able to stop the rents in the bottom with temporary plating. In some of the holds even the leaks were not closed, and the vessel was floated without a bottom. Pressure men, that remarkable class of men who make it their business to work in compressed air, and who are commonly known as "sand hogs," were brought from the Quebec Bridge, the caisson work of which had shortly been completed, or from New York, the superintendent of the work having for many years been engaged in compressed-air work about New York.

A wooden tank of about 200 tons capacity was built directly between the engines, and the weight of the engines (180 tons) was carried by this tank. On the day of flotation about twenty-five tones of water were left in the tank. As the vessel rose and the engines settled to their old level, blocking was put between the tank and the deck over it, and this water pumped out, the surplus lift of the tank being transferred to the vessel. The heavy tides of the St. Lawrence, although the center of the vessel was flooded, lifted the end of the vessel, and the craft rose and fell with high tide, so that the engines rose and fell on some occasions fourteen inches.

Air bags and tested barrels were used in the after bunkers. The boilers were blown out, and air was applied to the forward bunkers, they, too, being treated as caissons.

Several of the holds where the bottom was not so badly destroyed that it was necessary to treat them as caissons were pumped out, eight and ten-inch centrifugal pumps being used for this purpose.

Owing to bad weather the tugs which had been lying alongside on November 16, the date set for flotation, had dropped down the river to a more sheltered position. As the tide rose the air compressors were set to work and the full power of the plant used in forcing air into the holds of the ship. Suddenly there was a movement in the great vessel as she lifted herself from the rock and a cheer went up from those on board. Five minutes later the *Bavarian* floated clear of Wye Rock in sixty feet of water, and was hauled to her anchors, which had been set off her port bow and quarter. After the first few minutes all apprehension that the vessel might turn turtle or that the air pressure would not hold the water back was dispelled. The *Bavarian* floated on an almost even keel and was shortly after towed by tugs to Quebec. The wrecking operations were under charge of William Wallace Wotherspoon, C.E., superintendent. who had entire charge of the wrecking operations inside the bulwarks of the vessel, and Mr. R. O. King, C.E., who was controlling engineer. Capt. W. Leslie, of Kingston, had charge of all nautical work.—"Scientific American."

CONTROLLING TORPEDOES BY WIRELESS TELEGRAPHY.

A torpedo-launching apparatus, which is operated by electric waves, has recently been tried in France, on the Mediterranean coast. It consists of two cylinders placed one above the other, the upper one acting as a float which holds the mast wire, while the lower cylinder contains the torpedo and the launching device. The method of carrying out the maneuvers of the apparatus from a distance by the electric waves has been devised by M. Devaux, who gives the following account of the

method. Up to the present the Hertz waves have been utilized generally for operating an electro-magnet, whose armature makes the well-known signals as in the Morse instrument, and this is the method used in wireless telegraphy. However, the movement of the armature can evidently be used to set free the force of another instrument, thus acting as a relay. Up to the present, the devices in use allowed of using one electro-magnet for operating two movements only, corresponding to the up or down position of the armature. To carry out a series of maneuvers which have no connection with each other, it is necessary to have as many electro-magnets as there are operations. This, though complicated, would be possible in the case of wire transmission, but it is not practicable for electric waves, seeing that as yet we are unable to separate the waves so as to have them act on different coherers located at the same point. We proposed the problem of setting in movement from a distance by electric waves, of a series of forces, acting in an order always variable and keeping independent of each other, and we devised a new method for the operating apparatus. This apparatus is used with each of the coherers, and can also be used with a wire-transmission system. In the latter case they need only one wire, using the earth as a return.

The present system consists, first of a distributing device passing over all the contact points from which start the different operating circuits, and second, a commutating switch which sends the current at the right moment when the distributor has reached the proper circuit. To carry this out, an electro-

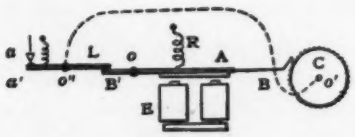


Fig. 1.

magnet E (Fig. 1) can attract an armature, A, which is held back by the spring, R, and is pivoted at O. This armature is prolonged at each end by the arms B and $B.'$ The arm, A,

provided with a pawl, works upon the ratchet wheel, *C*, which is mounted on the shaft, *O'* of the distributor. The wheel thus advances one tooth at each attraction of the electro-magnet. At the other end, the arm *B'* strikes each time against the end of the lever, *L*, which is pivoted at *O''* and which works the contacts, *a a'*. Mounted on the shaft, *O'*, of the distributor is an arm, *D* (Fig. 2), whose end works upon a set of

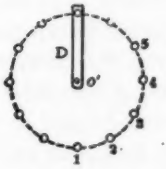


Fig. 2.

contacts 1, 2, 3, etc., and sends current from *O'* into twelve circuits. Each movement of the electro-magnet thus moves the arm around one contact. The electro-magnet is operated by aerial waves, using a coherer, or again by a line wire. In Fig. 3 is seen the second part of the device

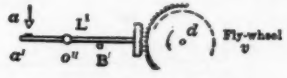


Fig. 3.

for shifting the circuits. The arm, *L'*, lies in another plane from the arm *B B'*, and on the forward end it carries a small rack and pinion device which is not seen in Fig. 1. The rack, sliding over the wheel, *d*, engages with it when the lever. *L*, is out of its normal position. In coming back, the rack draws the wheel with it, but as the latter carries a flywheel on the shaft, this gives it a certain inertia and retards the fall of the lever. During the series of attractions of the armature, the contact at *a a'* is thus kept open, and it is only when the armature is stopped and therefore when the distributor arm has come to the proper point, that the contact is closed.

The apparatus works as follows: When at rest, the arm, *D*,

remains upon contact number twelve, which is a dead point. The lever closes the circuit at *a a'*. We now wish to close circuit number seven without interfering with the others. It suffices to send seven currents or seven sets of waves with the rhythm $1=t$. The electro-magnet will work seven times at this frequency and will make the ratchet wheel, *C*, advance by seven teeth, whereupon the arm, *D*, comes upon point number seven. But now the contact at *a a'* has remained open on account of the inertia, as we have seen, of wheel *d*, and it is only closed when the arm, *D*, comes to rest. We thus close circuit number seven without operating the others. Again, we may wish to close several circuits at the same time. To do this, the distributor must be free to move without breaking the circuit which has been closed. It suffices that circuits one, two, three...twelve be closed by locked relays, and the latter serve to operate the work circuits. These relays are closed and remain so when the current passes, and their opening depends upon a device connected with one of the contact blocks.

In Fig. 4 is shown the apparatus for operation at a distance. It is connected in the place of the usual Morse receiver for aerial telegraphy. We have applied this apparatus to operating

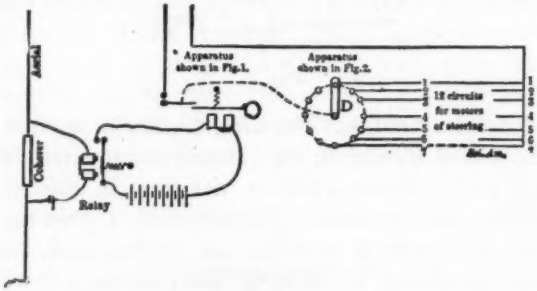


Fig. 4.

torpedoes at sea, and thus need the following maneuvers: 1. Forward run. 2. Back run. 3. Stop of propeller motor. 4. Rudder to right. 5. Rudder to left. 6. Stop of the steering motor. 7. Lighting of the signals at the front. 8. Lighting at rear. 9. Launching of the torpedo. The apparatus uses twelve points.

thus having three points free. The speed allowed for making a complete round of the distributor is two seconds. The nine working circuits closed upon seven locked relays which worked the propeller and steering motors, signals lamps and launching apparatus. Inside the torpedo is placed a set of Fulmen storage batteries which give a four-hours' run. The torpedo is formed of two cylinders of sheet iron with conical ends, connected one above the other. It weighs six and eight-tenths tons. The top cylinder, thirty feet long and two feet diameter, serves as a float, and it carries two small masts to which are fixed a receiving wire and a set of signal lamps. The lower cylinder, which is thirty-five feet long and three feet diameter, contains a torpedo-launching tube and a Whitehead torpedo, besides the battery, motors, etc. The land post has a mast wire fifty feet long, in fine strands. A distance from 400 to 5,000 feet was covered in the trial. The latter took place in the port of Antibes on the Mediterranean. These tests have proved satisfactory as regards the different maneuvers of the torpedo-launching apparatus from the post on shore.—"Scientific American."

HIGH DUTY METAL.

It is a well known fact that the copper alloys known as brass and bronze generally suffer a marked reduction in strength with increase of temperature. This loss of strength is not generally taken into account because the deterioration is slow up to a certain point, and this point is above the temperature of steam or air at the usual pressures. At about the temperature of 150 to 175 pounds steam the loss begins to be very rapid, and at about 400 degrees Fahrenheit there is a sudden marked drop in tenacity. From that point to 500 degrees Fahrenheit the loss is very rapid and the strength of the material is seriously affected.

In designing a new line of extra heavy and medium pressure brass valves, it was determined to use the metal which would show the smallest percentage of loss in this

respect that it was possible to obtain, the same metal to have other qualities which are essential in a valve at all times and particularly in a valve for use under high pressure.

In endeavoring to secure this metal, an effort was made through the Secretary of the American Society of Mechanical Engineers to collect all of the existing literature upon the subject, and it was discovered that practically all that has been determined on this question in the results of the experiments of the British Admiralty in 1877 is described on page 309 of Kent's Manual. Having found that no further light was to be obtained from past researches, there was inaugurated an exhaustive series of tests with a large number of mixtures of copper, tin, zinc and other metals, with the purpose of determining, absolutely, the best metal for the purpose.

These experiments have extended through a period of eight months, and the statements made above have been fully corroborated and emphasized. For instance, it was found that an alloy that is very commonly in use as steam metal and would be called a fairly good metal for this purpose, showed a drop in tensile strength of as much as 28 per cent. when raised to the temperature of 407 degrees Fahrenheit, which is about the temperature of steam under 250 pounds pressure without superheating. It was assumed that this temperature would be a fair one at which to make these tests, inasmuch as engineering practice is tending strongly in these days to the use of steam at about this pressure, or superheated from a lower pressure to this temperature.

Another metal which has been considered an excellent mixture and is frequently used by valve makers for valves of higher grade and designed for higher pressure, showed twenty-two per cent. loss under the same conditions.

The well known "government" mixture, as it has been called, consisting of eighty-eight parts of copper, ten of tin and two of zinc, was found to be as little affected by this extraordinary increase of temperature as any alloy which has ever been used, as far as known, in the manufacture of valves. The "government" mixture was found to have as an average of a large

number of bars tested, a cold tensile strength of 33,633 pounds per square inch. When raised to 407 degrees Fahrenheit and again tested, the temperature being maintained constant during the test, the tensile strength dropped to 30,675 pounds per square inch, showing a loss of nearly nine per cent.

We are able to say that, after making all of our experiments, we have arrived at an alloy which is practically of the same tensile strength as the above mixture when cold, as it shows an ultimate strength of 33,520 pounds per square inch at seventy degrees Fahrenheit, and further, it shows an ultimate strength of 31,627 pounds per square inch at 407 degrees Fahrenheit, the loss being only five and six-tenths per cent.

A table is appended which shows in summarized form the results of these experiments upon six different alloys, as follows:

	Tensile strength at 70° F.	Tensile Strength at 407° F.	Loss per cent.
No. 1	21,790	15,640	28.2
No. 2	29,010	22,410	22.4
No. 3	24,510	22,059	9.5
No. 4	33,633	30,675	8.8
No. 5	33,710	31,305	7.1
No. 6	33,520	31,627	5.6

Number one is the steam metal alluded to above and which is in common use among valve manufacturers.

Number two is a metal which we ourselves use at times and is fairly strong and durable at comparatively low temperatures.

Number three represents one of the earlier experiments illustrating the advance along the line of research, but indicates too low tensile strength.

Number four is the "government" mixture, so called.

Number five is one of the later attempts, and number six is the mixture which has finally been adopted and will be called by us "High Duty" metal.

In addition to the rare quality of maintaining the high tensile strength at high temperatures, this metal also shows

wearing qualities which are very remarkable; and as wearing qualities in the moving parts of a valve are of the utmost importance, the "government" mixture was again taken as the unit or standard, and it was found, by means of a special machine in which the movements of the parts of a valve were imitated exactly under extreme conditions of friction and at high temperature, that this metal wore away at about one-third the rate shown by the "government" mixture mentioned above.

This alloy has also been found to be very tough, resisting shock or water hammer with greater success, and is, in fact, far less brittle than any other of the metals tested.

In strength under compression it again showed marked superiority, as the flow of the metal was extremely uniform under high pressures, much more so than any other alloy upon which these tests were made.

Further, and this is very important, this metal makes sound, tight castings. The tests prove this conclusively.

WESTERN TUBE CO.

THE PRINCIPLES OF SIMILITUDE.

Some thirty years ago, or more, the late Professor James Thomson—who, in his own line, was as original and profound a thinker as his more famous, but younger, brother, Lord Kelvin—contributed to the Proceedings of the Institution of Engineers and Shipbuilders of Scotland a paper on the "Comparison of Similar Structures as to Elasticity, Strength and Stability." In this paper a number of very simple, but important, relations were established between the relative security of two structures differing from each other only in the matter of scale. Some eight years ago (see "Engineering," vol. lxviii., page 475) Professor Barr discussed the same subject with considerable wealth of illustration, and finally, in a paper read lately before the Institution of Automobile Engineers, 1, Albemarle Street, W., Mr. F. W. Lanchester, whose originality and success as a builder of motor cars of the highest class is well established, set forth once more these "Laws of Simili-

tude." In all these papers it has been shown that a model if made of the same material as its prototype will carry with the same factor of safety an equal load per square foot, if similarly distributed. At the same time the weight of the model varies obviously as the cube of the ratio of the scales, so that, so far as the stresses in it arise from the weight of its constituent parts, it is much less severely stressed. It is, therefore, quite safe to build a bridge or a roof by scaling down from a larger one designed to carry the same load per square foot. The stresses arising from the traveling load, or from wind pressure, will be identical in the reduced model, whilst the dead-load stresses will be less. Where, however, questions of stability rather than of strength are concerned, this process of scaling down is not free from danger, since the stability of a chimney, for instance, against overturning from a given wind-pressure per square foot is double that of another built to half scale. Hence, if Nature worked on strictly mathematical lines, a little woman should have disproportionately big feet as compared with her taller sisters.

The same theory of similitude shows that in a model engine the stresses which arise from a given steam pressure are exactly the same as in the larger original; and, further, that, so far as the inertia stresses are concerned, these will be the same in the two, if the piston speeds are the same. It follows, therefore, that the model can be run safely at a higher rate of revolution than the larger engine. The horsepower, developed is obviously proportional to the volume of steam passed per minute; that is to say, to $n\ d^2\ .\ s$, where n denotes the number of revolutions per minute, d the diameter of cylinder, and s the stroke. Using the small letters for the model, and large ones for the full-sized engine, we have, if the piston speed is the same, $n\ s = N\ S$, so that the horsepower of the model varies as $N\ S\ .\ d^2$, whilst its weight will vary as d^3. Hence the horsepower per unit of weight will, in the case of the model, be proportional to $\frac{N\ S}{d}$, whilst that of the larger engine will vary

as $\frac{N\,S}{D}$; and as D is greater than d, much more power could obviously be obtained from a given weight of metal by using it to build a number of small motors than to construct one large one. In fact, the weight of the smaller engine per horsepower developed should vary directly as the scale of reduction. No doubt it is to this fact that a portion of the enormous saving in weight shown in Mr. McKechnie's recent paper before the Institute of Naval Architects, to follow on the substitution of gas for steam in the propulsion of war vessels, is to be attributed. He proposed, it will be remembered, the construction of a cruiser, in which the 16,000 horsepower required to drive her were developed by gas engines, with forty cylinders in all, driving four shafts. With steam this power would be developed by, say, two four-cylinder engines driving two shafts; and were the gas engines arranged in similar fashion, the principle of similitude shows that their weight would be increased in the ratio $\sqrt{\frac{40}{8}}$, or be more than doubled. The weight of the producers would not, of course, be affected, but the modification here suggested would undoubtedly go far to cancel a large proportion of the great gain in weight claimed.

This conclusion from the principle of similitude may of course be challenged on the perfectly correct contention that in practice the law in question is more commonly violated than confirmed, and it is no doubt for this reason that engineers have made so little use of it. The late Mr. Normand was, however, a firm believer in both its validity and its utility, and on several occasions constructed marine engines of different sizes from the same set of drawings, merely using a different scale. In one instance (see "Engineering, Vol. LX, page 58) the ratio of the scales of the two engines was as much as one to one and nine-tenths. If, however, comparison is made of the weights and outputs of motors of different kinds made in a large range of standard sizes by good firms, not the slightest sign of any agreement between the horsepower actually developed per unit of weight in the different sizes and that theoret-

ically due is in general visible. Thus, from the catalogue of a firm making a popular type of high-speed engines in sizes ranging from 250 up to about 1,700 horsepower, the ratio of dimensions and the weights per horsepower corrected to the same piston speed are as follow:

Ratio of cylinder diameters.....	0.42	0.54	0.66	0.80	1
Total weight per B.H.P., pounds..	65	66	64	62	57

Again, from the catalogue of a large turbine maker we get the following figures:

Ratio of inlet diameters..	0.092	0.202	0.356	0.536	0.735	1.0
Weight in pounds per B.H.P. at 16-ft. head...	52.0	31.5	36.2	42.0	38.1	52.9

So that in both cases the weights per brake horsepower hour, instead of being less in the smaller sizes in the ratio of the corresponding dimensions, as theory indicates they should be, are actually more or less constant. Quite similar results would be obtained on making a similar comparison for a series of gas engines of the same type made by a single firm.

The reasons for this extraordinary discrepancy between theory and actual fact appear to be mainly that a firm's standard practice as to the minimum thicknesses of castings and the like is based upon their heaviest and most important work. In getting out a set of standard proportions for a series of engines the basis is the largest engine of the group; but whilst, as has been shown, the smaller sizes would be amply strong enough if simply scaled down from the larger drawings, this is practically never done, but the minimum scantlings used are not vastly different throughout the whole range.

It follows, therefore, that a small engine designed by a draughtsman accustomed to heavy work is unduly massive, though it generally proves reliable and satisfactory in working. The saving of a few pounds of cheap metal in a small engine is not felt to be worth the trouble of training the foundry to produce light castings, or the mental worry of departing from established practice in minimum dimensions. The difficulty a draughtsman of this kind has in "scaling down" is best ex-

hibited perhaps when he is set to tackle instrument design. The earliest of the cyclometers put on the market, for example, was designed by a capable mechanical draughtsman, who made a thoroughly "substantial engineering job" of it, with the result that the weight of the instrument ran into pounds, and its cost was something between a guinea and thirty shillings. The modern cyclometers, on the other hand, are the product of experience in instrument work. They are quite reliable, though their weight is a mere ounce or so, and their cost in pence much the same as that of their predecessor was when expressed in shillings. Of course, the scaling-down process advocated by Mr. Normand requires to be applied with discretion. Sound castings cannot be obtained below certain minimum thicknesses, and if scaling down leads to less scantlings than these, the design must be correspondingly modified. Moreover, all structures and machines have to be capable of withstanding, in addition to their intended stresses, the effect of accidental shocks and blows. These, in the case of large engines, may be insignificant compared with the regular working loads; whilst in a much reduced model they may give rise to stresses and strains much in excess of those legitimately obtaining.

Owing to the unconscious bias of the designer to base his minimum dimensions on the heaviest class of work with which he has experience, the figures tabulated above cannot be considered as proof that the principles of similitude are inapplicable in practice. As already stated, Mr. Normand utilized them boldly, and if instead of comparing together the different sized productions of one firm, those of different firms, one of which specializes in heavy work and the other in smaller engines, are contrasted, there appears to be a very fair agreement between the theory and actual practice. From the paper on "Recent Trials of Machinery of Warships," contributed to the Proceedings of the Institution of Civil Engineers by Sir A. J. Durston, K.C.B., and Mr. J. Oram, we take the following particulars as to the engines of the *Powerful* (cruiser) and the *Bullfinch* (torpedo-boat destroyer):

.....	Diameter of low-pressure cylinder.	Piston speed per minute	Steam pressure at engines.	Maximum horse-power.	Weight. H.P.
	inches.	*feet.*	*pounds.*		
Powerful.....	76	906	207	25,900	93
Bullfinch.....	34	1,200	210	5,800	22

Reduced to the same piston speed, the weight of the *Powerful's* engines would be 77.5 pounds per horsepower, so that the ratio between the specific weights is 3.15; whilst if the materials and design were the same in the two cases, this ratio should by the theory be that of the low-pressure cylinders, or 2.24. In this case, therefore, part of the large reduction in the specific weight is undoubtedly due to the difference in the design and the materials.

Similarly, if, instead of comparing the specific weights of a range of gas engines constructed by one maker, the specific weight of his largest, and presumably most carefully proportioned, engine is compared with a motor-car engine, the difference is extraordinary. Here again the matter is complicated by differences in design and in the nature of the materials used; but the economy of weight shown is far more than can thus be accounted for. The effect of an improvement in the quality of materials in augmenting the horsepower obtainable from a cylinder of given capacity was discussed by Mr. Lanchester in his paper already referred to. He showed, for example, that a ten per cent. increase in the tensile strength of the metal used for the reciprocating parts of a given engine would allow of an increase of fifteen per cent. in the power developed. He also generalized the usual statement that the horsepower of similar engines running at the same piston speed should vary as the square of the linear dimensions by showing that a more complete statement of the facts of the case is that it should vary as $D^n S^{2-n}$, where D is the diameter of the cylinder, and S the stroke. By then investigating the relation of the reciprocating stresses to the diameter and stroke of the cylinder, he arrived at the conclusion that $n=1.5$, and therefore proposed that in

rating petrol engines in hill-climbing contests and the like, the power should be taken as equal to $0.4.D^{1.5}S^{0.5}$. With this formula, he contended, extravagant freak designs, built merely to "beat the rating," will be eliminated.

The laws of similitude, as set forth above, are incomplete in the regard that no allowance has been made for the fact that, in general, a large engine is somewhat more efficient than a small one. In the case of a steam engine, for instance, the initial condensation is increased the larger the proportion of the surface exposed to the volume of the clearances; and, consequently, to obtain the same mean pressure in a model as in a larger engine, a greater quantity of steam must be passed into the cylinders, so that if the port area bears the same proportion to cylinder area in the small as in the large sizes, there will be a greater loss from wire drawing, and to obtain the same mean pressure a little later cut-off will be required. In the case of internal-combustion engines the loss to the walls is similarly proportionately greater in small than in large engines, since the ratio of surface to volume increases as the scale of the machine is diminished. The mean effective pressure in such engines is therefore somewhat less in small than in large gas engines, though a certain compensation is possible by the adoption of higher compressions and relatively longer strokes in the smaller sizes. The Crossley engine at the suction gas trials at Derby, for instance, had a cylinder 8½ inches in diameter and a 21-inch stroke, with a compression of 210 pounds per square inch. Large engines are not built with anything like this ratio of stroke to diameter, nor to work with such a high compression; about 140 pounds being the maximum with which these large engines can be worked on a commercial basis with producer gas, though the attempt has frequently been made to increase this figure. These considerations therefore place a limit to the increase in the efficiency of a gas engine with increase in size. Most makers agree that a 60-horsepower gas engine will be practically as economical of fuel as a larger one. A complete theory of similitude

should take into account the whole of these factors, but the requisite data for effecting this in a thoroughly rational manner are as yet wanting.—"Engineering," May 10, 1907.

JAPANESE GOVERNMENT STEEL WORKS.

We have, from time to time, noted the progress which was being made with the Imperial Japanese Government Steel Works at Wakamatsu, as the matter had special interest to the steel makers of this country. That progress has been interrupted on account both of financial and technical difficulties, and before the war with Russia the prospects of the undertaking were not by any means bright. A national struggle, such as that in which Japan was engaged, gives an impetus to all the industrial undertakings which are necessary for carrying it on; and the Government Steel Works seem to have been helped over their chief difficulties, and are now in a position to do very effective work. A very complete report of their present condition has been received by the Board of Trade from H.M. Chargé d'Affaires at Tokio, which contains some points of interest. After sketching the history of the undertaking, the most of which we have noted on previous occasions, the writer of the report reviews the expenditure, and makes out that the total of the sums appropriated for the establishment of the works amounted to nearly £2,000,000.

The area of the works is about 330 acres, including some 82 acres of ground recently purchased for the purpose of enlargement and not yet built upon. The works are quite close to Wakamatsu, the chief port for the export of Kyushu coal, and about nine miles west of Moji, the well-known coaling port on the Shimonoseki Straits, and northern terminus of the Kyushu Railway. The position was chosen largely on account of its proximity to the Chikuho coal fields, by far the most extensive coal-producing district at present known in Japan. This district lies some thirty miles to the south of Wakamatsu, in the provinces of Chikuzen and Buzen, and covers an area of over 320 square miles. The entrance to Wakamatsu Harbor is very narrow, opening to a basin about

a mile across at its widest part. This basin, again, opens to a large lagoon some ten miles in circumference. It is on the eastern side of this lagoon that the Imperial Steel Works stand. The lagoon is naturally very shallow, but dredging operations have been carried on since 1890 by the Wakamatsu Harbor Improvement Works, with the result that the depth of water alongside the quay wall for about 630 feet is twenty feet at ebb tide. A branch line of the Kyushu Railway has a station immediately adjoining the steel works, and short lines connect the various parts of the works and the quay (amounting in all to thirty miles of lines) with the Kyushu Railway. Coal can be carried by the branch line from the mines at Futase, in the Province of Chikuzen, which belong to the steel works. The present output from these mines is not sufficient to meet the demands of the works, supplementary supplies being obtained from private-owned collieries at Miike, in Chikuzen Province and the Island of Takashima, near Nagasaki. The total consumption of coal per annum is about 500,000 tons. A very complete coal-washing plant is in operation to supply the coke ovens. The cost of coke delivered at the blast furnaces is about twenty shillings per ton. A by-product plant is being built to take ammonium sulphate and tar from the Semet-Solvay Battery.

There are two blast furnaces in working order, and one in course of construction, which will be completed this year. The two furnaces now in use produce 300 tons of pig iron in twenty-four hours, one giving 175 tons and the other 125 tons. At present the blast machinery is driven by steam generated in twenty-four Lancashire boilers by the gas from the furnace tops, but a by-product plant is in course of construction for purifying the gas from the furnace tops, and using it in engines to drive the blast machinery.

The ore used in the furnaces is hematite, with some magnetite and limonite. About eighty per cent. of this ore comes from the Dayeh mines, near Hankow, in China, under special contract with the Hang Yang Iron Works, owners of the mines. The ore contains on an average sixty per cent. of iron, and at least 100,000 tons must be supplied annually. The cost of this

ore at the mines is from 4s. to 7s. per ton, and the freight to Wakamatsu is 7s. per ton, so that the cost of the ore delivered at the furnaces is from 11s. to 14s. per ton. An irregular supply of hematite ore is obtained from Korea, and considerable quantities from different parts of Japan; but the point of importance to be noted is that for purposes of manufacture the coal and iron fields of China are more convenient to the works than those of Japan, and this opens up great possibilities for the future.

There are three principal departments in the works, viz: (1) the pig-iron department; (2) the steel department; and (3) the rolling-mill department. Besides these there are the electric central building, central pumping station, iron foundry, repairing shop, pattern shop and foundry sand storage, boiler shop, smithy, chemical and mechanical laboratory and inspection bureau, and fire-brick plant. The buildings are lighted throughout by electric light.

There are at present two Bessemer converters, with a capacity of 150 tons each per twenty-four hours, one charge amounting to 10 tons. In three years' time a third plant will be completed, according to designs drawn up by the German expert in charge of the Bessemer department. There are now eight Siemens-Martin furnaces, with a capacity of about the same quantity of molten steel per twenty-four hours as the two Bessemer converters, *i. e.*, 300 tons. A great part of this steel is taken in 5-ton ingots direct to the plate mill. Under present conditions the works are able to turn out about 90,000 tons of finished material a year. The original plans were for an annual output of 60,000 tons, which would have satisfied one-half of the demands of that time, but the success of the venture, and the steady increase of Government requirements, have brought about a sensible extension of the original programme. In the course of the next five or six years it is confidently expected that the annual output will amount to 180,000 tons, *i. e.*, double the present output. By far the greatest portion of the products goes to the Imperial Navy Department, the remainder being purchased by the War and Railway Departments. Materials used at the various arsenals in Japan—Tokio, Kure, Sasebo,

Osaka, Yokiska and Maizuru—are largely supplied by the Imperial Steel Works. Practically all the materials for the the building of ships of war are now turned out at the works; it should be noted, however, that armor plate is not made there.

For the Railway Department 40,000 tons of rails are turned out yearly. These rails vary in weight from 9 pounds to 60 pounds per yard, the output of heavy rails being ten times that of light rails. Heavy rails are exported in large quantities to Korea for use on the Korean and connecting systems. The selling price of heavy rails is about £7, 10s. per ton. The prices paid by the three Government Departments for goods purchased from the Imperial Steel Works are not dependent on the current foreign market prices, but are arranged in advance on the basis of the average prices ruling abroad during the immediately preceding five years, due consideration being also given to existing conditions at the works.

The report to which we have referred gives a table showing the rolled products of the Imperial Steel Works. These include all the ordinary forms in use, and the results of the tensile tests show that the quality of the material is very good.

A large proportion of the machinery employed is of German make. In the first instance all machinery was procured from Germany with the exception of some electric cranes of American make, but now England and America are fairly well represented. The pig-iron and steel plant is practically all from Germany. The number of skilled and unskilled workmen employed at the works is about 7,000, with 3,000 coolies, bringing the total number of employees up to 10,000. The daily wage paid varies from 4½d. to 3s. 10d.—"Engineering."

SCREW PROPELLERS.

Read before the Liverpool Engineering Society by T. SIDNEY COCKRILL, M. I., Mech. E.

The object of this paper is to consider the general features of screw propulsion and propeller design, with which, no doubt, many members of this society are already well acquainted, and to bring to notice a few points bearing on the

subject which have occurred to me from time to time. There is no subject connected with marine engineering on which a greater diversity of opinion exists. Eminent members of the profession have, at times, laid down the most conflicting pronouncements with respect to screw propulsion. So it is hoped that this paper will lead to an interesting discussion, from which the author hopes to gather much useful information. The existence of such difference of opinion on the subject is probably due to the fact that little is really positively known of the mechanics of screw propulsion, or the forces set up by the action of the screw. There are so many disturbing factors, such as progression of the wake, and variation of its speed at different points, augmentation of the ship's resistance caused by the propeller itself, skin friction of the blades, effects due to rotation of the race, effects due to position of the propeller, eddies, cavitation and numerous other influences whose effects cannot be calculated, that it seems almost hopeless to attempt to use first principles in solving the problems involved.

The want of exact scientific knowledge is to be deplored, for when one contemplates the very extensive use of the screw propeller, and how largely the speed and economy of the ship depend on its design and proportions, it is seen that the saving which would follow on more perfect knowledge must be very great.

To meet this want, the governments of this and other countries, and one or two shipbuilding companies, have constructed experimental tanks in which the performances of modern propellers and ships of varying forms can be ascertained; and the results arrived at have been found so closely to agree with the actual performances of the full-size propellers and ships, the dimensions and forms of which have been determined by the model experiments, that there is no doubt but that in the future tank experiments will become the practice for determining the most suitable dimensions of propellers. At the present time, when the turbine, with its increased speed of rotation, is replacing to such an extent the

reciprocating engine for vessels of over 15 or 16 knots, modifications in propeller design render previous practice almost worthless as a basis for designing new propellers, and the experimental tank is practically the only means of securing the highest possible efficiency.

The greater part of our knowledge of screw propellers is due to the two Froudes, father and son, who carried out a series of experiments in the Admiralty tanks, extending over some 30 to 40 years.

The principle on which the screw (as well as all other marine propellers) acts, is that the reaction caused by the projection of a stream of water by the propeller in one direction produces motion of the vessel in the opposite direction. And since rate of change of momentum is proportional to the force which causes it, the force exerted by the propeller is proportional to the momentum per second of the stream of water which it projects. Action and reaction are equal and opposite, and therefore the force exerted by the propeller in driving the water astern, is equal to the thrust in the ahead direction; and this again is equal to the resistance of the ship. Using standard units of measurements, this means, in short, that the momentum of the stream projected astern by the screw is the measure of the resistance of the ship.

The momentum of the stream, or "race," as it is called, is the product of the mass of the water projected in one second and its velocity relatively to the surrounding water, or $M = m(v - V)$. And the mass of the race projected in one second in turn depends on the area of its cross section and its velocity relatively to the ship $m = \frac{W}{g} Av$. The momentum of the race is, therefore, equal to $\frac{W}{g} Av(v - V)$ and this is the measure of the thrust of the screw, and also of the resistance of the ship.

The energy expended in setting in motion the water forming the race is lost so far as propelling the vessel is concerned.

This energy is $= \frac{W}{2g} Av\ (v - V)^2$. That is to say, it varies directly as the mass of the water acted upon, but also as the square of its final velocity. If the mass of the water acted upon were doubled, the loss of energy would be doubled; but if the final velocity were doubled, the loss of energy would be increased fourfold. The most efficient propeller is, therefore, that which projects the greatest mass of water astern at the lowest speed.

Theoretically, the lower the sternward speed of the race relatively to the ship v, for a given value of V, the greater is the efficiency; and it follows that the larger A is made, the more efficient would be the performance.

As the sectional area of the steam of water A which is projected astern by the screw type of propeller is, owing to practical circumstances, larger than that projected by any other type, the velocity of the stream v may be less with the screw than with other types; and this is really the fundamental reason why the screw is the most efficient form of propeller. There is, of course, practically only one other type of marine propeller now used—the paddle wheel; but, in spite of certain disadvantages, the screw has proved to be the better type, taking everything into consideration, and is used very much the more extensively.

The formula for the exact value of the thrust of a given screw propeller, even when it is working under the most favorable conditions, has yet to be invented. The leading principles involved in screw propulsion may, however, be defined.

The resistance of the ship, after its inertia has been overcome, is due to the resistance of the water and the resistance of the air. The resistance of the water may be sub-divided into frictional, wave-making, stream-line and eddy-making resistances.

The resistance in a sea-way is further augmented by the wind and waves, and the pitching of the ship. The resistance in smooth water can be estimated with a very small degree of error, but that in a sea-way is beyond the range of mathe-

matics. It may, however, be remarked that length, size and weight in a ship seem to counteract the adverse effects of wind and waves in a sea-way; and we in Liverpool can bear witness to this, having become quite accustomed to the regularity with which our large ocean liners arrive in port under all conditions of weather.

Referring to the disturbing influences acting between the thrust of the screw and the resistance of the ship, it is proposed at this point to consider two of the most important, one of which, it will be seen, is due to the effect of the presence of the ship on the propeller, and the other to the effect of the propeller on the ship. The first is the wake, or the forward motion imparted to the water through which the vessel moves; and the second is the augmentation of the ship's resistance caused by the sucking action of the propeller.

When a vessel moves through the water it tends to drag with it the water immediately surrounding it. This is caused chiefly by the friction between the skin of the ship and the particles of water in contact with it, but also by the stream-line motion and the waves set up by the motion of the vessel. The water which thus tends to accompany the ship is technically known as the wake. The speed of the wake is greater in the immediate vicinity of the sides and bottom of the ship, and becomes gradually less the further it is removed. It is also greater at the after part of the vessel, on account of the motion being gradually impressed on the water as the vessel moves through it; and for the same reason the speed of the wake is greater in the case of long vessels.

The water in which the propeller works is, therefore, not at rest, and the speed of the propeller through the water is not the same as the speed of the ship. If we denote the speed of the ship by V, the speed of the wake by w, and let V_1 equal the speed of the propeller through the water in which it works, then $V_1 = V - w$. That is to say, the propeller advances through the water at a speed *less* than that of the ship by an amount equal to the speed of the wake. The effect of this is that the thrust of the propeller is *greater* than it would be

were it working in still water. The increase in thrust due to the wake means that a portion of that part of the energy developed by the engines which is expended in giving motion to the wake, is returned as useful work in the form of an augmentation to the natural thrust of the propeller.

In twin-screw vessels, owing to the propellers being further removed from the hull, the increase in thrust derived from the wake is less than that in the case of single-screw ships.

The second disturbing influence is the effect the propeller has on the ship behind which it works. When a propeller is revolving and throwing a stream of water in a sternward direction, it has the effect of reducing the pressure of the water in front of it. Owing to this, the pressure of water on the after part of the ship is less than it would be if the propeller were at rest. There is therefore a reduction in the water pressure behind the ship, and the effect of this on her performance is precisely equivalent to an increase of pressure in front, viz.: an augmentation of the resistance against which the ship moves. The propeller has therefore not only to develop an amount of thrust to overcome the natural resistance of the ship, or, in other words, to exert a force equal in amount to that which would be required to *tow* the vessel at the same speed, but it has to exert an additional amount of thrust to overcome that augmentation in the ship's resistance which is due to the action of the propeller itself.

Here again, as in the case of the effect of the wake, there is a difference between twin-screw vessels and single-screw vessels. Twin screws, being further removed from the hull, do not cause so much loss in pressure on the after part of the ship as single screws, and therefore the augmentation of resistance is less.

Coarse-pitch propellers produce a greater augmentation of resistance than fine-pitch propellers; this, perhaps, accounts for the higher efficiency of fine-pitch propellers.

Considering the net result of the gain in thrust resulting from the wake conjointly with the loss due to the propeller augmenting the ship's resistance, it was found by the Froudes

that for most types of vessels the gain due to the wake and the loss due to the augmentation of resistance so nearly balance one another that they may be neglected in most calculations affecting propeller design. The cancellation of these two factors (each of which, taken by itself, would be sufficient to upset any prediction as to performance) is due to the fact that for most types of ships, the conditions which are favorable to causing an increase in gain of thrust from the wake, are also those which tend to an increase in the augmentation of resistance; and *vice versâ*. It must not be forgotten, however, that in the case of vessels of unusual proportions of speed, or other conditions affecting the design of the propeller, it is unsafe to take it for granted that these influences are in equilibrium.

NUMBER OF SCREWS.

In order to obtain the required thrust to overcome the total resistance, it is necessary to have a certain propeller-disc area if the slip of the propeller is to be kept within reasonable limits. The disc area of a propeller is the area of a circle of the same diameter as the diameter of the propeller over the blade tips. To obtain this disc area it may be necessary to have only one propeller; or two, or even three or four propellers may be required. It is the disc area on which the area of section of the stream thrown sternwards depends, the stream being, in fact, an annular column equal in area to the disc area less the area of the boss. In the case of cargo steamers of moderate speed and ordinary draught of water, the thrust required is easily obtained from one screw. The draught of water in these vessels not being restricted, a propeller of large diameter may be used, giving the required area of section of stream without resorting to duplication of screws.

With higher speeds the resistance increases quickly, being as the square of the speed for moderate speeds; but the resistance often varies at a very much higher power of the speed at higher rates of speed; so that in the case of passenger vessels it is necessary, as a rule, to employ twin screws to obtain the necessary thrust.

When the draught of water is limited again, twin screws may be necessary, on account of the draught being insufficient to allow of a single screw of sufficient disc area and adequate immersion.

In the case of very large powers it is deemed inadvisable to transmit the whole power through one line of shafting, the diameter of shaft required being too large to ensure sound forging. And, again, twin screws with two sets of engines obviate to some extent the risk of total disablement at sea.

In the British Navy twin screws are the rule, chiefly on account of there being less chance of total disablement in action, but also on account of greater maneuvering ability (which is, of course, a very important consideration in warships), and on account of reduction in the height of the engines, whereby the cylinder tops are well below the armored or protected deck, so minimizing the risk of injury from shot and shell. In the American and German Navies many of the battleships have triple screws driven by three sets of engines, the center engine and propeller alone being used when steaming at cruising speeds.

The practice of fitting two or three propellers on one shaft for high-speed vessels has, it seems, been abandoned, probably on account of the interference which must inevitably take place. It is noteworthy that in the case of some high-speed yessels (the *Turbinia*, for instance), which had originally two or three propellers on each shaft, a single propeller of larger diameter has been fitted to each shaft, and better results have been obtained.

NUMBER OF BLADES.

There is not much doubt but that a propeller with two blades is more efficient under favorable conditions than propellers with three or four blades, but its efficiency is greatly impaired in a sea-way. The higher efficiency of the two-bladed propeller is due to absence of interference between the blades, and to there being less edgeway resistance. Experience has taught us, however, that on the whole, in actual practice three blades are best for propellers which can be well

immersed below the surface of the water, such as the relatively smaller screws of vessels with two or more screws; and that four blades are best for the large propellers necessary for single-screw ships, these being comparatively near the surface of the water.

SHAPE OF BLADE.

In designing the shape of the blade, it should be borne in mind that whereas blades having their greatest breadth comparatively near the tip, develop more thrust, yet this advantage is practically nullified by the greater frictional resistance of blades of this shape, owing to the broadest part of the blade being situated near the part of greatest circumferential velocity.

The number of patents taken out for shapes of propeller blades alone is legion, but experience seems to prove that the elliptical shape is preferable for most ordinary cases. The Admiralty have long adopted this shape for all classes of warships, and there seems to be no reason in the majority of cases for departing from it.

Speaking generally, the shape of the blade has little influence on propulsive efficiency, but vibration may be reduced by having a suitable shape. The chief considerations as regards efficiency are the relations between diameter, pitch and surface. It is more than probable that the shape of the section of the blade is of greater importance than the shape of the blade itself. The blade should be as thin as possible consistent with strength, so as to reduce its edgeway resistance as it revolves in the water. This rule applies with increasing force from the root towards the tip, the circumferential velocity being higher as the tip is approached, and the resistance increasing proportionately as the square and still higher powers of velocity.

The edges of the blade should be fine and sharp, and the shape of section of the blade should conform to the natural stream lines for the speed intended, so as to prevent eddies and reduce edgeway resistance to a minimum.

The blades of high-speed vessels are usually made of bronze,

owing to the strength of this material allowing of very thin blades, and to its being non-corrosive; the blades have knife-edges, and are polished on both sides.

The use of bronze for propeller blades for all services should, the author thinks, have more consideration than it appears to have at present. Bronze blades may be made much thinner than cast-iron blades, they have smoother surfaces, the edges preserve their keenness, and they do not deteriorate by corrosion and pitting, so that the resistance of the propeller itself is very much reduced; and, after all, there is the intrinsic value of the bronze as scrap to be taken into account when considering the increased first cost.

DIAMETER OF PROPELLER.

The diameter of the propeller is fixed by the required disc area, about which something has already been said when considering the number of screws. Theoretically, the efficiency rises with increase in the weight of water acted on, so that the larger the diameter the higher will be the efficiency. The diameter should never be so large, however, as to cause the tips of the blades to break through the surface of the water, or even to be near the surface under ordinary trim, as then the blade carries air down with it, and the presence of air in the race is the cause of a great loss in efficiency. The mixture of water and air is not only of lower density, but the presence of air lowers the speed at which water will flow to the propeller, and the result is racing or acceleration of the speed of rotation. Racing in a sea-way is, the author believes, as much due to the presence of air in the race as to the blades being partly out of the water.

A circumstance which tends to upset the theoretical rule that the larger the propeller the higher will be the efficiency, is that the resistance of the blades increases more and more rapidly with each unit of increase in diameter. A large-diameter propeller, while being efficient owing to the large amount of water acted upon, may absorb so much power by its own resistance that it would not give such good results as

a propeller of smaller diameter. The point is, in fixing the diameter, to ascertain at what point in the scale of diameters the rapidly-rising curve of resistance overtakes the less rapidly rising curve of theoretical efficiency. The efficiency of reciprocating engines, however, usually rises with decreased revolutions, so that even the "ultimate" efficiency of the propeller may be sacrificed to some extent in order to gain in the total efficiency of the engine and propeller combined.

Too small a diameter results in an abnormal amount of slip and consequent waste of power.

Vessels with bluff lines require larger-diameter propellers than vessels with fine lines, the water in which the propeller works being more disturbed. With a propeller of large diameter the blades reach out into more solid water.

Referring to propellers driven by turbine engines, the thrust developed is proportional to the square of the diameter, D^2, and also to the square of the velocity of the stream driven sternward, presumably $= (P \times R)^2$. These terms are reciprocals of one another to obtain the same thrust. But if the diameter be made too small, and the $P \times R$ too great, a loss in efficiency will result, owing to an abnormal slip. In the engines the velocity of the turbine blades is fixed almost beyond consideration, being about 0.5 that of the steam flowing through the turbine. The mean velocity of the blades $V = D_2\ \pi\ R$; where $D_2 =$ the diameter of the rotor taken half way along the blades. It will be seen that an increase in revolutions R means a decrease in the diameter of the rotor D_2, and consequent saving in the weight and space occupied by the turbine. It is therefore very desirable to run the turbine at as high a speed of rotation as possible, but as this also means reduced diameter of propeller and consequent loss in propeller efficiency, a compromise must be made so that the combined engine and propeller efficiency may be as high as possible. It appears probable that a higher combined efficiency would result if slightly larger turbines than those in use at present, with corresponding decrease in revolutions and increase in propeller diameter, were adopted.

The point to be inculcated with regard to propeller efficiency is the absolute necessity of a free and unrestricted flow of water to the propeller; and this can only be obtained by molding the run of the vessel as fine as possible, and of such lines as not to interfere with the natural stream-like motion of the water as it closes in under the stern.

It may be remarked here that all obstructions to the free flow of water round the propeller boss should be avoided, especially for high speeds. The nuts for securing loose blades should be in recesses, with plates fitted over, to preserve the contour of the boss. A conical cap should always be fitted at the after end of the boss. The length and shape of this cap should be designed to suit the stream-lines abaft the boss for the intended speed, for if the cap does not *entirely* fill the cavity formed by the water behind the boss it will not serve its purpose, and may just as well not be fitted. Means should also be provided for preventing eddies at the after ends of the stern tubes and other places, and for preventing the too sudden parting of the water at the fore end of the propeller boss.

BLADE AREA.

The area of the blades, or blade surface, is conveniently expressed as a fraction of the disc area.

The following table gives the ratio of blade area to disc area which is generally adopted:

Cargo vessels with full run, single screw, . . .	.31 to .34
Cargo vessels with medium run, single screw, .	.34 to .37
Cargo vessels with fine run, single screw, . . .	.37 to .39
Launches, single screw, ...	.34 to .36
Fast passenger vessels, twin screws, ...	.40 to .43

It is generally thought inadvisable to go beyond .5 or .6 on account of causing interference between the blades; but in some high-speed turbine vessels the ratio is considerably higher than this.

PITCH.

The pitch is sometimes made to increase gradually from the root to the tip of the blade; and in some propellers the pitch is uniform over the blade except near the boss, where it is finer. The object of a decreased pitch near the boss is to minimize the churning of the water which is supposed to take place at that part.

Some makers, on the other hand, cause the pitch to increase *across* the blade—finer for the leading or forward half of the blade, and coarser for the following or after half. The idea in this is that the water driven astern is gradually accelerated. This can only be recommended in the case of very wide blades with high pitch ratio.

Experience shows more and more, however, that little advantage, if any, is gained by a variable pitch; and uniform pitch is becoming the standard practice with most makers. The Admiralty seldom adopt any kind of variable pitch.

SLIP.

The slip is the difference between the speed of advance of the propeller (supposing it to be working in an unyielding substance) and the actual speed of the ship. In other words, it is equal to the pitch multiplied by the revolutions, less the distance traversed by the ship. If the water did not yield to the propeller and flow sternward, the speed of the ship would be the same as the speed of the propeller, and there would be no such thing as slip; but water, being a fluid, is driven astern by the action of the propeller as the ship moves ahead. The rate at which the water is driven astern relatively to the surrounding water is usually said to be equal to the slip; but this is only true provided the pitch multiplied by the revolutions is equal to the speed of the race relatively to the ship, or, in other words, provided the propeller itself does not slip in the race. As far as the author can see, we have no means of ascertaining the truth of this.

The above, however, only relates to *apparent* slip, for it does not take account of the fact that the propeller is not

working in still water, but in water in motion in a forward direction owing to the influence of the ship in passing through it; and, as it is the *propeller* that is under consideration, the speed of the propeller and not the speed of the ship through the water should be the basis in calculating the slip.

For example, if the pitch were 15 feet, the revolutions 81 per minute, and the speed of ship 11 knots: Then

Speed of propeller $= \frac{P \times R \times 60}{6{,}080} = \frac{15 \times 81 \times 60}{6{,}080}$ = 12 knots

Speed of ship, ... = 11 knots

Apparent slip of propeller, ... = 1 knots

But supposing the wake to have a speed of 2 knots: Then

Speed of propeller as before $= \frac{P \times R \times 60}{6{,}080}$ . . = 12 knots

Advance of propeller through the water in which it works $= V - w = 11 - 2$, ... = 9 knots

"Real slip" of propeller, ... = 3 knots

It is sometimes found that a propeller seems to have *negative* slip, that is to say, that the speed of the ship is (apparently) greater than the speed of the propeller which drives it. This negative slip is only found when the speed of the ship is taken instead of the speed of the propeller through the water; it is the apparent slip, not the real slip, of the propeller.

The phenomenon of negative slip is the cause of a vast amount of ingenious, if not always scientific, hypotheses to account for its existence. Everyone agrees that negative real slip is a physical impossibility. In the author's humble opinion negative apparent slip is also a physical impossibility, and therefore there is no need to account for a thing which does not exist. This is on the understanding that "apparent slip" means $\frac{P \times R}{60} - V$ while "real slip" means $\frac{P \times R}{60} - (V - w)$; these are the usual definitions of the terms, and they are given here to prevent misunderstanding arising from the mere

meaning of words. The explanation is simple enough, for in order to calculate the slip, the pitch P must first be known; it is one of the factors in the calculation, and, in the author's opinion, no one knows what the true effective pitch of a propeller is. What is usually taken as the pitch of a propeller is the pitch of the after face of the blade, or, rather, the mean of the various pitches found at different parts of the after faces of the blades. This is certainly not the effective pitch of the propeller with respect to its action on the water, one reason at least for this being that it takes no account of the curved back of the blade, and it is certain that this increases the true effective pitch, although no one knows exactly, or even approximately, what value is to be assigned to it. There are also other reasons for supposing that we do not know what the effective pitch of a propeller is.

The cause of negative slip often given is that the propeller is working in the wake of the ship, and therefore working in water which has a forward motion.

For example of this reasoning we may take a previous example. Supposing the speed of the ship to be 13 knots and the speed of the wake 4 knots.

Then speed of propeller, as before, . . .	12 knots
Speed of ship, ...	13 knots
Apparent slip of propeller, ...	−1 knot
And speed of propeller, as before, . . .	12 knots
Advance of propeller through the water = 13−4 =	9 knots
"Real slip" of propeller, ...	3 knots

This reasoning seems to take it that the forward motion of the wake is greater than the backward motion imparted to the water by the propeller, so that the water forming the race still has a slight forward motion after it has been acted upon by the propeller. We would then have a large volume of water (the wake) following the ship, *part* of that volume (the race) is *impeded* only (not even brought to rest) by the pro-

peller. The net result of the wake and the race would be motion in the *same* direction as the motion of the ship. But the motion of the ship is due to re-action on the water, and therefore the net result must be water flowing sternward. The author thinks there is no doubt whatever about this, and that any theory which supposes that the net result on the water of a self-propelled vessel is motion of water in the same direction as the motion of the vessel, is condemned from the outset.

An explanation of negative slip put forward by Mr. Barnaby is based on the fact that the feed water to the propeller is accelerated while still forward of the propeller; and the action of the propeller upon the water while in contact is to accumulate pressure, which has the effect of increasing the acceleration of the race after it has left the propeller; and that, owing to this, the slip or acceleration of the water in the race is always in excess of the slip of the screw as given by its "nominal" pitch. But it rather seems to the author that this not only explains negative slip, but it explains it away, for it means that what we call the pitch of the propeller is not the true effective pitch.

In connection with slip there is a paradox which should be noticed. It is quite true that the most efficient propeller is that which projects the greatest mass of water astern at the lowest speed. This is equivalent to saying that the most efficient propeller is that which has the least slip. But we know by experience that a very small amount of slip, or no apparent slip, or negative apparent slip, is a sure sign of inefficiency; and that a propeller giving no slip, or negative slip, should be changed as soon as possible, as it is not suited to the vessel. How are these apparently contradictory statements to be reconciled? Both are true; but while in the first statement efficiency is taken in a general theoretical sense, neglecting friction, or, in such a sense, for instance, as we should use if we were comparing the theoretical efficiency of screws and paddle wheels, or screw propulsion and jet propulsion, or even screw propulsion itself applied to different vessels, in the second

statement efficiency is taken with respect to the actual results (which include friction and a host of other unknown and incalculable factors) of a particular case. The explanation stated less generally is this: Theoretically, a propeller having a small amount of slip is efficient; but smallness of slip is invariably due to fineness of pitch; and the finer the pitch the greater the loss by surface friction, until a point is reached at which the loss by friction overcomes the gain in the theoretical efficiency, and any further reduction of pitch produces a loss in what may be called the actual efficiency.

CAVITATION.

In very high-speed vessels such as destroyers, the troublesome phenomenon termed "cavitation" is likely to occur if the blade surface be too small for the thrust developed. Cavitation is failure in the supply of water to the propeller, and results in the formation of a vacuum or cavity at the forward faces of the blades owing to the water being unable to follow the blades fast enough. It was first discovered by Mr. Barnaby during the trials of H. M. destroyer *Daring* in 1894. It may be likened to what would occur in the case of a pump plunger which rises so quickly that it exerts a pull on the water beneath it of more than 15 pounds per square inch. In that case the pressure of the atmosphere is insufficient to cause the water to follow the plunger, and a vacuum or cavity is formed. In the case of propellers a cavity is formed on the forward side of the blade if the peripheral speed is too high, on account of the head of water being insufficient to cause it to follow the blade; and the consequence is a reduced thrust.

At the surface of the water, cavitation would take place when the pull of the blade on the water exceeds the pressure of the atmosphere; and below the surface, when the pull exceeds the pressure of the atmosphere plus the pressure due to the head of water. This, however, leaves out of consideration, for the sake of simplicity, the fact that, owing to the water giving off vapor when the pressure is reduced, cavitation actually takes place when the pull is somewhat less than this.

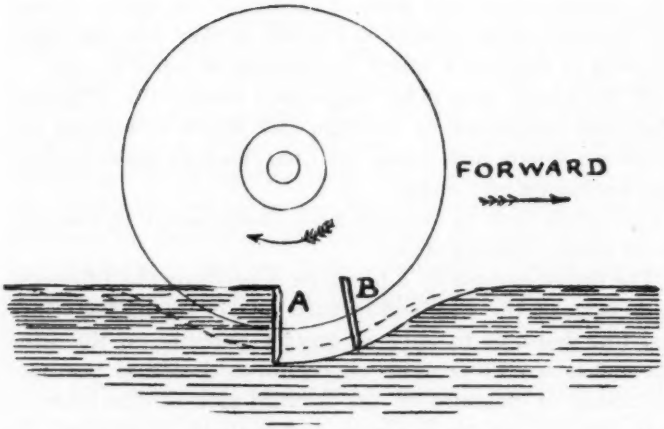
It commences to take place at the tips of the blades (where the circumferential velocity is highest) as they pass the highest point of their path where the pressure of water is least.

In destroyers and other high-speed vessels the projected blade area should not be less than that which will insure the thrust per square inch being less than about 12 pounds, otherwise cavitation may ensue.

Cavitation may be detected by a sharp ascent in the slip curve above a certain speed.

The thrust exerted by a propeller depends on the difference in pressure on the after and forward sides of the blades. It is therefore desirable to have as great a pressure on the after side and as small a pressure on the forward side as possible. Reduction of pressure on the forward side of the blade is equivalent to increase of thrust, and is to be sought for. It would therefore seem, on the face of it, that if the propeller revolved so rapidly that the water was unable to follow the blades, so causing the pressure on the forward side to fall to zero, a highly satisfactory state of things would be arrived at. But further consideration will show that if the water does not follow the blades the pressure on the after side will also fall, as there would be a lack of water there also, so causing a greatly diminished thrust. It is a *sine qua non* that to obtain efficiency the water must be in contact with the forward side of the blade.

Cavitation is more easily grasped by considering the paddle-wheel. Referring to the sketch, which shows a hypothetical case for the sake of simplicity we will suppose we are dealing with only two floats A and B. The float A enters the water so rapidly that a cavity is formed forward of it. So far as this float is concerned, a highly satisfactory state of things is arrived at; for we have a great pressure of water on the after face of the float and no pressure on the forward face, tending to create a great amount of thrust to drive the vessel ahead. But it is apparent that the float B has no water to engage with, and a loss in thrust is therefore inevitable. (What would actually happen in a real case, with the full number of



floats and the vessel moving ahead, is that the water line in the vicinity of the wheel would probably take the form shown by the dotted line, that is to say, all the floats would be only partially but equally immersed.)

Almost the same sort of thing happens to the screw propeller; you cannot have great pressure on the aftersides of the blades while there is a cavity on the forward sides, owing to an insufficient quantity of water flowing past them.

Cavitation is due to the slip of the propeller; if there were no slip there could be no cavitation. This will be more easily seen by referring to the sketch of the paddle wheel. In this paper the author has touched upon a few of the leading features only of screw propellers and propulsion; to deal with the subject at all fully would be impossible in a short paper. It is the most recondite and difficult subject that marine engineers have to deal with with, and it gives rise to the greatest difference of opinion. But the author feels sure that, in time, the apparent vagaries of the screw will be entirely mastered by theory and reduced to mathematical formulae; and then we may expect marine engineering to progress by leaps and bounds.

SHIPS.

UNITED STATES.

Viper—*Launch of.*—The submarine *Viper* (*No. 10*) was successfully launched March 30, 1907, at the works of the Fore River Shipbuilding Co., Quincy, Mass.

Tarantula—*Launch of.*—The Submarine boat *No. 12*, the *Tarantula,* was launched at the works of the Fore River Shipbuilding Co., March 30, 1907. This is the last of the four submarines building by the Electric Boat Company to be launched.

Birmingham.—The Scout Cruiser *Birmingham* was successfully launched at the works of the Fore River Shipbuilding Co., Quincy, Mass., on May 29.

The *Birmingham* is one of the three scouts authorized by Act of Congress, approved April 27, 1904. The other two are the *Salem* and the *Chester.* The *Salem* is also being built by the Fore River Shipbuilding Co., while the *Chester* is building at the Bath Iron Works, Bath, Me.

The following are the chief characteristics of the *Birmingham.* Displacement, 3,750 tons; length, 423 feet; beam, 46.6 feet; draught, 16¾ feet; I.H.P., 16,000; armament, twelve 3-inch guns; torpedo tubes, 2; speed, 24 knots; coal, 1,250 tons.

The propelling engines will be right and left, placed in watertight compartments and separated by an athwart-ship bulkhead. They will be of the vertical, inverted-cylinder, direct-acting, triple-expansion type. The general particulars of engines will be as follows:

Diameter of high-pressure cylinder, inches	28½
intermediate-pressure cylinder, inches	45
forward low-pressure cylinder, inches	62
after low-pressure cylinder, inches	62
Stroke of all pistons, inches	36
Indicated horsepower of both main engines	16,000
Revolutions	200
Steam pressure, at high-pressure receiver, pounds	250

The main valves will be worked by Stephenson link motions with double-bar links. There will be one piston valve for the high-pressure cylinder and two each for the intermediate and low-pressure cylinders.

The framing of the engines will consist of forged-steel columns trussed by forged-steel stays. The engine bed plates will be of cast steel, supported on the keelson plates. All crank, thrust, line and propeller shafting will be hollow. The shafts, piston rods, connecting rods and working parts generally will be forged steel as specified.

Each main condenser will have a cooling surface of about 8,000 square feet, measured on the outside of the tubes, the water passing through the tubes.

The main circulating pumps will be of the centrifugal type, one for each main condenser. For each main condenser there will be an independent air pump of the double, vertical, double-acting type.

There will be an auxiliary condenser, of about 600 square feet cooling surface, in the forward engine room, connecting with all the auxiliary machinery.

There will be an auxiliary condenser of about 200 square feet cooling surface in the dynamo room for the exclusive use of the dynamo engines.

The auxiliary and dynamo condensers will have air and circulating pumps.

There will be twelve water-tube boilers of the "Express" type, placed in three watertight compartments, as shown.

The total grate surface will be at least 693 square feet.

The total heating surface will be at least 37,080 square feet.

The forced-draft system will consist of blowers discharging into air-tight firerooms, there being two blowers for each boiler compartment, as shown.

There will be four smoke pipes, as shown.

There will be a main feed pump, a fire and bilge pump, and a bilge pump driven from the forward end of the crank shaft in each engine room.

There will be a feed pump and a fire and bilge pump in each boiler room.

The starboard propeller will be right-hand and the port propeller left-hand, the engines turning outward when going ahead.

There will be ash hoists, ash ejectors, turning engines, auxiliary pumps, refrigerating machinery and motors or steam engines, workshop machinery and motors, distilling and evaporating apparatus, steam reversing gear with oil-control cylinder, and such other machinery, tools, instruments or apparatus as are hereinafter described, together with such other as may be necessary to complete the machinery in accordance with the specifications and plans as finally adopted.

All parts of machinery and boilers will be secured in an approved manner to prevent displacement when the vessel is used for ramming.

Pipes, gear and other fittings will be so arranged that the tubes of boilers, condensers, evaporators, feed heaters, distillers, etc., may be withdrawn without interference, and so that parts of main and auxiliary machinery may be easily removed for examination and repair.

See JOURNAL A. S. N. E., Vol. XVII, p. 263, for a full description of this vessel.

Submarine Tests.—*Octopus and Lake.*—Interesting trials between the submarine boats *Lake,* built by the Lake Torpedo-boat Co., of Bridgeport, Conn., and the *Octopus,* representing the Holland type of boat, built by the Electric Boat Company, of New York, began at Newport, R. I., May 1. The *Octopus* was the first boat to start on a test on May 1. She had to abandon the trial, however, because she broke a bracket of her port engine just as she was about to be sent over the measured-mile course. Before the accident the *Octopus* had several successful unofficial runs over the course. The members of the naval trial board were aboard the *Octopus* when the bracket broke.

The *Lake* underwent her first official test on May 2, which she completed without any mishap, in the most successful manner, with the members of the trial board aboard. The conditions of this day's test were that the boat should make three

surface runs under both her motors and her gasoline engines and three under her gasoline engines alone. A strong northeast wind was blowing down the course and, with the tide setting in, the sea was as choppy as it ever is on the bay except in storms.

Her unofficial time is given as eight and six-tenths knots an hour. Under her gasoline engines alone the reduction was hardly one knot, the rate being 7.68 knots. It is said that tidal corrections will give the *Lake* an even better record, for the current which she faced for some time was strong and for a time she had to encounter a choppy sea.

The *Lake* reached the upper line of the course in 7 minutes and 16 seconds, a rate of 8.26 knots. Running back, the boat had the advantage of the wind, but the tide was against her and the sea rough. The *Lake*, however, made a fine run, her time being 6 minutes and 33 seconds, or 9.16 knots per hour. The average of these two runs was a rate of 8.71 knots. The run up the course again, which was against the wind, but with a little tide favoring the boat, was made in 7 minutes and 43 seconds, a rate of 7.83 knots. Down and up the course the average speed was 8.49 knots. This completed the try out of the boat at a maximum speed under both her motors and her gasoline engines, and it was found that the average rate was 8.6 knots an hour.

In the speed tests of the *Lake* under gasoline engines only the conditions of wind and sea had improved, and on the third run down the bay, with wind and tide with her, she made her best speed of the day, which was at the rate of 9.18 knots. Her first run under engines was made in 6 minutes and 57 seconds, a rate of 8.63 knots. Her second run, against both wind and tide, was her slowest, the time being 9 minutes and 18 seconds, and the rate 6.45 knots. The average rate of these two runs down and up the course was 7.54 knots. The mean average speed for the three runs with gasoline engines alone was 7.68 knots. Five more runs under the gasoline engines alone were made at gradually reduced speed to enable the Board to make a speed curve from which

to figure exact deductions. The *Octopus* having been repaired was to undergo her trials on May 2.

Advantage will be taken of our increased knowledge of submarine construction to so alter the *Pike* and *Grampus*, which have been so long at the Mare Island Yard, as to greatly increase their factor of safety. Larger batteries will lengthen their period of possible submersion to three hours, and bilge keels will enable the crew to move about more freely without altering the displacement of the vessel. Hooks for hauling the vessels to the surface in case of accident are to be placed on the side.

The *Octopus* in her first trial on May 3 broke two speed records while going over the Government's measured-mile course. In three times over the course the boat made an average speed of a fraction over eleven knots, while using only her gasoline engines. Besides being tested for surface runs, the *Octopus* had a series of trials in the semi-submerged condition, and in these made ten knots, with electric motors alone in use. To show the *Octopus*' ability to go under water quickly, the boat made a dive, going down twenty feet and immediately coming to the surface again.

When she had finished her first run and her speed had been checked up it was found that she had made the mile in 5 minutes 11 seconds, which was a rate of speed of 11.6 knots. Both down and up the course, or against and with the tide, the *Octopus* had made a rate of 10.78 knots, and her mean for the three runs at maximum speed as required by the specifications was a small fraction over eleven knots. Running semi-submerged, or awash, the *Octopus* gave another fine exhibition, attaining a maximum of 11.5 knots with a strong tide with her, and a mean of 9.98 knots.

More surface runs were made at gradually reduced speed to enable the Board to prepare a speed curve of the *Octopus*. The *Octopus* had a strong tide in her favor in making her first run awash, which was down the course. Only her electric motors were used, and her speed under the conditions was perhaps more surprising than it had been on the surface. The

mile was made in 5 minutes and 26 seconds, which was a rate of speed of 11.05 knots. Up the course the *Octopus* encountered a strong tide, but she was able to make the mile in 6 minutes and 41 seconds, the rate in knots being 8.97. The round trip both with and against the tide was made at a rate of just about 10 knots. Down the course again with the tide the *Octopus* made the run in 5 minutes and 29 seconds, a rate of 10.94 knots.

Her owners had claimed only 10.5 knots for her in cruising trim, and her performance gave a general surprise. Capt. Simon Lake, the inventor and the attachés of the *Lake,* were among the first to give their competitor credit for her record. Captain Lake is, however, greatly pleased with the performance of his own boat on May 2. He has never claimed great speed for her, but is relying on other advantages and facilities which have attracted attention.

Under perfect weather conditions the *Octopus* underwent her second test on May 8, and made what is claimed to be a world's record by attaining a speed of more than 10 knots under water. The best time heretofore made was 8½ knots by one of the submarines of the British Navy. The *Octopus* made three high-speed runs under water with Captain Marix on board. She glided through the water with apparent ease and covered the first mile at the rate of 10.15 knots. This was her maximum speed, while her average was 10 knots plus, which was only a fraction less than she made May 3, while running semi-submerged. According to the conditions the top of her conning tower had to be at least ten feet below the surface, which meant that those aboard would be at least twenty-three feet below. She had, however, to show at least three feet of a mast above the water as a marker for the observers, and from the top of this a little silk flag fluttered. After her submerged run the *Octopus* was put through several tests which came under the head of maneuvering or handling. The conditions require that the time be taken in diving to certain depths and returning to the surface, and in this series the *Octopus* made two porpoise dives. She went down at an eight

degree angle to a depth of 26 feet in a fraction less than 40 seconds. Then she returned to the surface so that an observation of 5 seconds could be made from her conning tower, and made another dive and another broach under the same conditions and in the same time. The best previous record for such diving was 46 seconds, made by the submarine boat *Fulton* of the *Octopus* type.

A feature of the tests was the use for the first time of the submarine bell, with which the *Octopus* is equipped. Another bell and receiving instrument were attached to the tender *Starling*, and communication was kept up all the time the submarine was under water.

The *Lake*, which was to have begun her submerged trials on May 8, sprung a leak in her torpedo tubes, and it was decided to postpone her trial until the damage was repaired.

In the trial of the submarines in Narragansett Bay, R. I., May 10, before the Navy Trial Board, the submarine boat *Octopus* established another record in making quick submergences while going through maneuvers in the government tests. The test was made in a gale blowing forty miles an hour. The boat, while going at full speed on the surface, twice shifted her motive power, adjusted her diving rudders and dived to a depth of twenty feet in an average time of 4 minutes and 30 seconds. The best time in which any submarine now in the Navy has ever gone through the same maneuvers is 11 minutes. The boat was also put through a series of turning trials, both on the surface and beneath, to determine her tactical diameter. The board tested the air compressors on the *Octopus*, which have to stand a pressure of 2,000 pounds to the square inch. They stood the test satisfactorily.

The *Octopus* had another test of endurance on May 11 in a ten-mile run while partially trimmed or in diving condition, and an average speed of 9.89 knots was attained on the entire run, which was made with the gasoline engine as a motive power. On this run the engines, according to the figures of the trial board, made forty-two revolutions more a minute than

the standardization figures called for. In trimming for the run the *Octopus* occupied a little over a minute.

Aside from the government test the *Octopus* had two extra tests at the request of the builders. These included six runs over the course, three under one engine, while the other was being used to charge the storage batteries, and the other three while the boat was being propelled by only one propeller. On both of these tests the boat made nearly eight knots.

A successful test was made on May 13 in connection with the trials of a submarine life preserver, cap and jacket. The apparatus resembles an ordinary diving suit, the air being supplied by a tank of oxylite contained on the inside of the jacket. A member of the *Octopus*' crew was submerged in one of the suits for four minutes. The *Octopus* made a run of thirty-five miles by storage-battery power to determine the distance she could go without recharging. According to the figures of the board, the boat could run submerged 115 miles with one charging of the batteries.

On May 14 the *Octopus* was subjected to various maneuvering and special trials and torpedo firing tests. Among the maneuvers were trials of the boat's ability to remain at rest while submerged, turning trials under various conditions and motive powers and quick stops. In the latter the boat came to a full stop from full speed ahead in 50 seconds. Of the special tests the most important was of the automatic devices for blowing out the water ballast to allow the boat to come to the surface in case of accident. In this test the *Octopus* came to the surface from a depth of 40 feet in 43 seconds. Only two torpedoes were fired. These were discharged as the boat was about to dive to a depth of 20 feet. An irregularity with the tubes prevented the discharging of the remaining two after the boat had submerged. Only one of the torpedoes was recovered, the other not coming to the surface.

Both the submarines *Octopus* and *Lake* began their twenty-four-hour submergence test on May 15 at the naval coaling station at Bradford in about 30 feet of water. The *Octopus* carried down a crew of sixteen men, while nine men went down

in the *Lake*. Both boats came up to the surface on the afternoon of May 16 after their twenty-four-hours' continuous submergence test, and none of the men on board of them looked any the worse for being bottled up under the sea for twenty-four hours. Incidentally new records of sustained life under the sea had been made, the seventeen hours for which a party on the submarine boat *Fulton* was submerged at New Suffolk, L. I., in the spring of 1902, having been the previous record. Later in 1902 a party lived twelve hours on the *Fulton* submerged at the torpedo station.

The scientific results of the interesting experiments will not be reduced to figures for several days. A sample of the air in each boat was taken in a bottle at stated periods, and this will be examined for carbonic acid gas and chloride gas, which are very noxious and may have been generated during the four hours the engines were under way.

Captain Cable, of the *Octopus*, said that he had used about one-forty-fourth of the stored air in his boat, from which he estimated that so far as air was concerned he could have remained below for three days longer. He said they ate and slept below just about as they would have done had they been in their homes. Captain Lake, of the submarine *Lake*, said: "We could easily have remained submerged for another twenty-four hours. We used a comparatively small supply of compressed air—in fact, not much more than two hundred pounds, and this was not applied until near the end of the test. If we had used oxone, of which we had plenty on board, and if we had been allowed to open our diving door to admit water to the diving compartment with which to force the carbonic acid gas, we would have drawn very little, if at all, upon our supply of compressed air. With all the *Lake's* various air-disposing arrangements in use a crew in the boat could easily stand a submergence of three days."

The submarine *Lake*, on May 17, went out on a series of maneuvers for three hours to prove the worth of her engines and motors and also her mobility under various conditions. Later in the day she began a thirty-hour endurance run in the

bay, and in all she gave an excellent account of herself. One of the *Lake's* reversing trials attracted much attention from the fact that the reverse was made by merely changing the pitch of her screws, which rendered stopping or reversing her engines unnecessary. Going ahead full speed on the surface under the power of both her engines she was reversed by this changing of pitch and became dead in the water in one minute and three seconds. As a result of her thirty-mile endurance run it was observed that her radius of action under her gasoline engines, carrying the normal amount of fuel, was about 440 miles. She cruised up and down the bay at the rate of about 7.25 knots, at which it took her four hours and thirteen and one-half minutes to cover the distance. Of her normal store of gasoline, which is about 1,500 gallons, she used about 110 gallons. She has emergency tanks, however, which enable her to carry about 4,500 gallons, with which her radius of action would, of course, be greatly increased. Her machinery stood the test of the long journey well.

The *Cuttlefish,* one of four submarines built at the Fore River Shipbuilding Company's yards, at Quincy, Mass., by the Electric Boat Company for the Government, was subjected to a 200-foot submergence test about six miles off Boston Light on Saturday afternoon, May 18, and the boat stood the test without damage or strain of any kind. No member of the crew went down in the boat, it being lowered below the surface by a large derrick vessel. Three submergences were at 150, 175 and 200 feet depths. When it was drawn up a very critical examination was made of it, and it was found that not a drop of water had leaked in anywhere, and that her machinery, which was carefully inspected, was in perfect condition. At a depth of 200 feet the pressure on the boat was about ninety pounds to the square inch, while on the whole boat there was a combined pressure of 15,000 tons. The *Cuttlefish* is the same type of boat as the *Octopus,* now under trials at Newport.

Trials with the *Octopus* were resumed on May 20 in Coddington Cove for torpedo practice. She carried four torpedoes and fired three at a target supposed to be a battleship,

marked by two rowboats 500 feet apart. The range was about 800 yards. The torpedoes were fired while the boat was running in a submerged condition, taking observations by the periscope and going at full speed. The first torpedo, fired from the port tube, ran straight to the target, then turned to a sharp angle to the right. The second got hung up in the starboard tube. The third torpedo went straight, but did not carry more than 500 yards. The *Octopus* also underwent anchor-gear trials and a successful test of her wireless apparatus. Later in the day, when the *Octopus* was at her slip at the torpedo station, she fired another torpedo from her starboard tube along the breakwater, and it went in good shape, apparently showing that there was something wrong with the torpedo previously fired from that tube. One of the torpedoes was lost, making the second lost by the *Octopus* within a week. They are valued at $3,000 each.

The submarine boat *Lake* had her submerged speed trial in Narragansett Bay May 21, and was also put through other maneuvering tests. Previous to the trial, the *Lake* and the *Octopus* were taken out to Brenton's Reef Lightship with the idea of giving them a sea trial in rough water. The wind was strong, but it was offshore, and the sea was comparatively smooth. The boats then returned to the harbor, and the *Lake* was sent over the measured-mile course three times. On the first run, with the tide and against the wind, she covered the course in 10 minutes and 29⅘ seconds, a speed of 7.72 knots an hour. The next run was made against tide and with the wind, and the *Lake* developed a speed of 5.69 knots an hour. On the third run against the tide the speed was 5.61 knots. The *Lake* was then sent over a course on which full speed was given her and then required to stop and back, a feat which she accomplished in 42 seconds. She made the maneuvers of sealing her top and dropping to twenty feet in 7 minutes and 19 seconds, and followed this by anchoring in forty feet of water, as required by the Board. The boat then returned to the harbor.

The *Lake* had her torpedo-firing trials and a run in the open sea on May 22. Her torpedo-launching outfit worked per-

fectly, each of the three projectiles being started off smoothly and, except in one instance where the discharge was evidently premature, the directions were accurate. The firing took place in Coddington Cove, two small boats being placed three hundred feet apart as a target. The target was made by two boats, anchored three hundred yards apart, to represent a battleship. The range was 800 yards. Observation of the target for the first two shots was made by the armiscope, while the *Lake* was submerged, but on the third shot observation was made from the conning tower. The first shot left the starboard tube all right, but the starting lever of the torpedo failed to trip. The second shot, fired from the port tube, was a success, the torpedo crossing the line between the two boats. Then the *Lake* started to turn to fire a torpedo from the stern tube, but the torpedo was discharged prematurely, and made only a short run through the water. All these torpedoes were later picked up, but there was no reloading on board. The *Lake* next executed some maneuvers at a depth of ten feet, completing a circle in 3 minutes and 35 seconds. Late in the afternoon the *Lake* was ordered to sea for a trial in light and awash conditions. The boat went through the test well, although the sea was running high.

The *Lake,* on May 23, gave a wonderful performance of submerging herself, and with her crew on board went five feet deeper than any boat has ever done, with human beings in her, thus establishing a new record. She went down 135 feet. The world's record previous to this was 130 feet, made by a French submarine. The *Lake's* pressure gauge at 125 feet showed fifty-two pounds to the square inch. It was 7 minutes and 15 seconds before her coning tower disappeared under the water, and from the time the *Lake* was thirty feet beneath the surface till she reached the bottom it was only 3 minutes and 45 seconds. In all it required the boat 19 minutes and 18 seconds from the time she was sealed to sink to the bottom, and altogether the boat was sealed about 20 minutes and 48 seconds. After she came to the surface a hurried examination showed that no water had leaked into the *Lake,* which was verified in a more careful examination that was made later. The boat's

machinery had also escaped damage, for under power of her gas engines she ran a mile into her slip at the torpedo station. A model of the Burger subsurface boat was also given a trial by the Board. She made three runs over the measured-mile course in the bay, the first at a rate of 8.93 knots, the second at a rate of 8.59 knots and the third at a rate of 9 knots, the mean being 8.78 knots. The boat is only 35 feet on the water line and is driven by a 28-horsepower gasoline engine.—"Army and Navy Journal."

ENGLAND.

Afridi.—The torpedo-boat destroyer *Afridi*, together with her sister ship *Ghurka*, will constitute the fastest class of vessel in the British Navy. The *Afridi* is an ocean-going torpedo-boat destroyer, was launched May 8th, from the Elswick shipyard of Sir W. G. Armstrong, Whitworth & Co., Limited. Her length between perpendiculars is 250 feet, and her breadth, molded, 25 feet. Her depth, molded, is 15 feet 6 inches, and her mean draught 7 feet 1 inch. The ship will carry three 12-pounder quick-firing guns, two of which will be mounted on the forecastle deck, and one on the upper deck aft. She will also be fitted with two 18-inch torpedo tubes, which will be mounted on the upper deck.

The machinery will be supplied by the Parsons Marine Steam Turbine Company, Limited, of Wallsend-on-Tyne. It will consist of a set of compound turbines, which will drive three propeller shafts, each fitted with one propeller. The boilers will be of the Yarrow type, and will develop approximately 14,500 horsepower. It is interesting to note that the Admiralty have decided to use oil instead of coal in these ships. Both the *Afridi* and the *Ghurka* have been designed for a speed of 33 knots.

Ariel.—During the maneuvers at Malta, April 19, the British destroyer *Ariel* struck the breakwater at the entrance of Grand Harbor, broke in two and sank. One man was drowned.

The Cossack, an ocean-going torpedo-boat destroyer, built by Cammell, Laird & Co., Birkenhead, was launched February 16th. Dimensions: Length, 270 feet; beam, 26 feet. She

has Parsons turbines, with triple screws. The designed speed is 33 knots.

Launch of a New Cruiser.— The first-class armored cruiser *Defence* was successfully launched at Pembroke Dockyard on April 27th. The new cruiser is a sister ship to the *Minotaur,* which was launched from Devonport Dockyard in June last, and her distinctive feature is her armament, which will be mounted entirely on the upper deck or above it. She will have four 9.2 inch breech-loading guns mounted in pairs in barbettes situated one on the forecastle deck and the other on the upper deck aft, and both at the middle line, and ten 7.5 inch breech-loading guns mounted five on each side in barbettes at regular distances apart over the portion of the ship occupied by the boiler and engine rooms. The mounting of so many guns on the upper deck of the *Defence* necessitated the reduction of weight in the upper structure of the ship, and, accordingly, instead of the vertical side armor being carried upwards from the lower to the upper deck in the central part of the vessel, it is stopped at the main deck. The side armor forms a complete belt round the ship, and is six inches thick outside the engine and boiler rooms, tapering to four inches towards the bow, and to three inches towards the stern. The barbettes are formed of armor seven inches thick, and in addition to the guns referred to, the *Defence* will carry sixteen smaller quick-firing weapons. She has also been fitted with five submerged torpedo tubes, and her engines are designed to give a speed of twenty-three knots per hour. The cost of the *Defence,* according to the latest revised estimate, will be £1,362,970.

Sale of Obsolete War Vessels. —Several obsolete war vessels have been disposed of at the following prices: Battleship *Sans Pariel,* T. W. Ward, Sheffield, £26,600; battleship *Conqueror,* Castles' Shipbreaking Company, Ltd., London, £16,800; armored cruiser *Undaunted,* Harris Bros., Bristol and Falmouth, £14,400; torpedo gunboat *Alarm,* the Shipbreaking Company, Ltd., London, £3,650; steam yacht *Ware,* W. Thomas and Son, shipbreakers, Anglesea, £925; torpedo-boat destroyer *Skate,* Cox & Co., Falmouth, £305.

Warships at present under Construction in the United Kingdom.

Nationality.	Description.	At Royal Dockyards.			At Private Yards.			Total.	
		Yard.	No.	Displ'm't.	Yard.	No.	Displ'm't.	No.	Displ'm't.
				Tons.			*Tons.*		*Tons.*
British	First class battleships	Portsmouth	1	18,600	Elswick	1	18,600	5	88,800
		Devonport	1	18,600	Dalmuir	1	16,500		
					Jarrow	1	16,500		
	First class armored cruisers	Chatham	1	14,600	Clydebank	1	17,250	7	109,100
		Devonport	1	14,600	Elswick	2	30,800		
		Pembroke	1	14,600	Govan	1	17,250		
	Torpedo-boat destroyers	....	....	....	Birkenhead..(2) Cowes.....(1) Elswick .....(1) Hebburn .....(1) Woolston ....(2)	7	6,585	7	6,585
	First class torpedo boats	....	....	....	Chiswick .....(2) Cowes.....(6) Hebburn.....(2) Poplar.....(3) Woolston.....(2)	15	3,725	15	3,725
	Submarine boats	Chatham	2	630	Barrow	12	3,720	14	4,350
	Total	....	7	81,630	....	41	130,930	48	212,560
Foreign or not stated	Battleships	....	....	....	Elswick	1	19,000	1	19,000
	Cruisers	....	....	....	Barrow	1	15,200	1	15,200
	Torpedo-boat destroyers	....	....	....	Jarrow	2	740	5	1,760
					Poplar	3	1,020		
	Total	....	....	....	....	7	35,960	7	35,960
Total		....	**7**	**81,630**	....	**48**	**166,890**	**55**	**248,520**

—"Lloyd's Quarterly Shipbuilding Return."

Launch of H. M. S. "Invincible."— The new cruiser *Invincible* was launched from the Elswick yard of Sir W. G. Armstrong, Whitworth & Co., Limited, April 13th. Complete success attended the event, which is as one would expect, inasmuch as there have already been floated from the same establishment, since it was first inaugurated 22½ years ago, twenty-four fighting ships for the British Navy, apart altogether from the work done for foreign Powers.

The *Invincible* excels, alike in length, speed, and gun power, any of her predecessors. She has a length of 530 feet, a beam of 78 feet 6 inches, and when displacing 17,250 tons draws 26 feet of water. At this draught she has a capacity for carrying 1,000 tons of coal, and a large quantity of oil fuel. While protected as effectively as the majority of battleships, excepting perhaps only the *King Edward VII* and *Dreadnought* classes, she is excelled in her gun power only by the *Dreadnought*. The main armament includes four pairs of 12-inch guns, mounted in four barbettes—one forward, one aft, and two amidships, the latter *en échelon* between the second and third funnels, which are, therefore, at a greater distance apart than the two forward funnels. The magazines for these guns are located between the second and third boiler-rooms. This arrangement of guns will enable the amidship guns to fire on either broadside, although their arc of training forward and abaft of the beam on the off broadside may be more limited than in the case of the *Dreadnought*. In the broadside action, however, the ship will have the same fire as the *Dreadnought*, while forward as well as aft she will use six 12-inch guns. The secondary armament for repelling torpedo attack consists of a large number of 4-inch quick-firing guns. The turbine machinery, which is being constructed by Messrs. Humphrys, Tennant & Co., of London, is to develop 41,000 indicated horsepower, and the legend speed is 25 knots. There are four screws and two rudders. All of the screws have four blades, and the outer propellers are about 30 feet ahead of the inner propellers, the latter being close up against

the rudders, which are entirely suspended from the counter of the ship within which the steering gear is fitted.

The vessel is in an advanced state, the launching weight being 8,400 tons. The drags consisted of nine piles of chains, resting on the ground on each side of the ship, the total weight being 450 tons. The cradle, which in the case of fine-ended ships is always a subject of interest, was made up somewhat differently from the practice in most naval construction works. The rows of poppets at the forward end were not continuous. There were three spaces left. The tops of the poppets were housed in the steel structure, from which there were plates extending under the ship, from the port to the starboard poppets; but instead of this metal structure, with its lashings, being secured to the hull, as in some instances, it was independent, and the ship was snugly cradled inside the plating with timber wedges. In this way some play was possible within the cradle, a matter of some importance when the stern of the ship is floated in the process of launching. It has been found in recent launching practice that, in cases where the framing has been bossed out for twin-screw propeller shafts, the stern becomes waterborne at an earlier period than was formerly the case, and, as a result, the consequent thrust on the forward cradle is earlier and the tendency to movement of the ship of greater duration. There is, however, the further fact that the speed down the ways does not attain the same high velocity. In the case of the *Invincible* the time occupied from the first perceptible movement until the vessel was completely afloat was just under 50 seconds. It follows that the distance which the vessel traveled from the end of the ways is shorter.

The release of a vessel of such weight is also of interest, and the method adopted at the Elswick Works was notably successful. The last dagger dropped was hydraulically operated. This dagger was of steel, pivoted to the bottom of the standing-way, and projecting through it into a bearing constructed of forged steel in the bottom of the sliding-way. To the upper part there was secured a counterbalance weight,

while the bottom abutted against a ram working in a cylinder in which the hydraulic pressure was 100 pounds. Lady Allendale, who released and named the ship, working a small capstan, opened the valve, and the receding of the arm within the cylinder caused the dagger to incline to the horizontal position, and thus gradually released the sliding ways and the ship. The ship was held rigidly until the moment of release, and at the same time there was certainty of release. Indeed, the whole performance was characterized by precision; and at the luncheon there was general congratulation upon the success of the day's proceedings, and upon the prosperity of the Elswick Works under Sir Andrew Noble, who presided.—"Engineering."

Launch of H. M. S. "Indomitable."—On Saturday, March 16th, his Majesty's ship *Indomitable* was successfully launched into the Clyde by the Fairfield Shipbuilding and Engineering Company, Limited, Govan. The launch was to have taken place at a quarter past two o'clock, but the tide proving especially good, it was found expedient and advisable to float the vessel a little before the pre-arranged time, and in this way a goodly proportion of the intending spectators were disappointed in not witnessing the actual send-off. The religious service usual at the launch of vessels for the British Navy was conducted on the special platform at the bow of the ship by the Rev. Rodger S. Kirkpatrick, minister of Govan Parish, and at four minutes past two o'clock the Marchioness of Breadaldane, who had been requested by the Admiralty to perform the ceremony of naming and releasing the vessel, cut, with an ornamental hatchet, the cord which controlled the arrangements of release. The customary bottle of wine was smashed against the ram—a feature which is much less pronounced than in older types of warships—and a moment later the vessel began to move down the ways. From the first movement until the hull became completely waterborne the time occupied was 1 minute 20 seconds. The tugs in attendance smartly took the *Indomitable* in charge, and she was soon berthed in the company's fitting-out basin.

On account of the Admiralty having made strict stipulations to that effect, no official description of this latest and mightiest of British cruisers has been furnished by the Fairfield Company. So faithfully has this desire for secrecy been regarded that amongst the workmen, and, indeed, all interested in naval matters on the Clyde as well as over a wider area, the *Indomitable* has come to be spoken of as "the mystery ship." The class of which she is a unit has excited world-wide interest, and although the details of design have not been officially disclosed, the principal particulars are not unknown to those well informed on naval subjects. With due reservation, what follows regarding the Fairfield cruiser and her two sister ships—the *Inflexible*, on the stock at Clydebank, to be launched on March 30th, and the *Invincible*, being built at Elswick, to be launched on April 13th, may be taken as in the main correct.

The dimensions, power and speed, etc., of the three ships, which are identical in design, are: Length, 530 feet; breadth, 78 feet 6 inches; draught, 26 feet; weight of hull, 9,660 tons; displacement, 17,250 tons; indicated horsepower, 41,000; speed, 25 knots; coal capacity at load draught, 1,000 tons. The vessels have a complete broadside armor from bow to stern, the maximum thickness being 7 inches, tapering to 4 inches at each end. The armor extends from under the water line to near the upper deck. In respect of protection, therefore, the *Indomitable* and her consorts are, perhaps, superior to previous vessels, just as they are certainly superior to their predecessors in gun power. Indeed, this new class of cruiser might, almost under any condition, tackle heavy battleships. Their length is 30 feet greater than that of any preceding British cruiser, and the displacement tonnage is 2.650 tons more than that of any hitherto built. The Fairfield-built cruiser *Cochrane*, for instance, which has recently been put through all her trials with conspicuous success, is 480 feet long as against the *Indomitable's* 530 feet, has 13,550 tons displacement as compared with the 17,250 tons, and 23,500 indicated horsepower as compared with 41,000 indicated horse-

power. The designed speed of the *Cochrane* was 23 knots as compared with the 25 knots of the *Indomitable.*

The machinery of the three vessels is of the Parsons steam-turbine type, steam being supplied from water-tube boilers, some of the vessels having the Yarrow boiler, and others the Babcock & Wilcox. As in the case of the battleship *Dreadnought,* the new cruisers have four lines of shafting, each carrying a three-bladed propeller of bronze. In the disposition of propellers, and in the design of the after run of the vessel, it was seen, from distant observation of the vessel before her send-off, that considerable departure from conventional practice has been made. The general impression conveyed was of effort and ingenuity having been exercised in diminishing needless excrescence and surface in the form of "deadwood," and leaving everything conducive to solid and undisturbed water in the region of the propellers; the rudder apparently being hung or supported on a very light structural arrangement. On each of the two inner shafts there will be a cruising turbine, a low-pressure ahead and a low-pressure astern turbine, while on each of the outer shafts there will be a high-pressure astern turbine, as well, of course, as the forward turbine. For low powers the steam thus passes through three turbines before entering the condenser, and the range of expansion is suggested by the fact that the blades of the turbines vary in length from something less than 1 inch up to 16 inches. There is a longitudinal bulkhead dividing the port and starboard engine rooms, in each of which there are thus five turbines and a condenser.

Even more than speed, the dominant feature of the *Indomitable* and her consorts is armament. The advance in this respect over previous vessels of the kind is enormous. Compared, for example, with the first-class cruiser *Cochrane,* recently handed over by Fairfield, the *Indomitable* has over double the hitting energy. The *Cochrane* has six 9.2-inch and four 7.5-inch guns, equal to a force of 161,470 foot-tons for one full broadside round. In the *Indomitable* there are eight 12-inch guns, equal to the enormous destructive power

of 381,576 foot-tons for one broadside round. This forms a striking example of the advance made within a period of two years in the aggressive or defensive powers of a type of vessel in which at first high speed was the distinctive quality aimed at. In the new vessels there is a corresponding gain in bow and stern fire, as they can operate six 12-inch guns ahead and six astern.

The progress made with these ships has been very satisfactory. The keel of the Fairfield vessel was laid on March 1st, 1906, and by the end of this March she will have had spent upon her the sum of £1,470,375. The keel of the Barrow ship, the *Inflexible*, was laid on February 5, 1906, and has had voted for progress with her construction up till the end of March the sum of £1,421,479. The keel of the *Invincible* at Elswick was laid on April 2d, and on her there will have been spent up to the end of March, £1,417,732. There is every prospect of all the vessels being completed for service fourteen or fifteen months hence, well within the contract time of two years and three months.—"The Engineer."

New Destroyers.—There was successfully launched from the works of Messrs. John Samuel White & Co., Ltd., East Cowes, Isle of Wight, on March 15, H. M. S. *Mohawk*, one of the new ocean-going destroyers building under the naval program of 1905–6. Her length is 270 feet, with a displacement of 800 tons. She will have an armament consisting of three 12-pounder quick-firing guns, and three 18-inch revolving torpedo tubes. The speed, to be maintained on a six-hours' full-power trial, is 33 knots, the radius of action at economical speed being 1,500 nautical miles. The vessel will be propelled by turbine machinery comprising five turbines (three ahead and two astern), driving three shafts and propellers, built by Messrs. J. S. White & Co., Ltd., under license from The Parsons Marine Steam Turbine Company, Ltd. The power of the machinery aggregates about 14,500 I.H.P. Steam will be supplied by six boilers, each of about 2,400 H.P., of the "White-Forster" type, made by the same firm, these boilers being fired by liquid fuel on a system which has been

experimented with successfully by the British Admiralty for some two or three years. No coal stowage is provided in the vessel, and she will rely entirely on the liquid-fuel installation. The air pumps, forced-lubrication pumps, boiler-feed pumps, oil-fuel pumps, fire and bilge pumps, and distilling plant are being supplied by Messrs. G. &. J. Weir, Ltd., Glasgow, the fans and circulating engines by Messrs. W. H. Allen, Son & Co., Ltd., of Bedford, and the air compressing machinery and electric light plant by Mr. Peter Brotherhood, Westminster Bridge, London, S. E.

The Torpedo-boat "Grasshopper."—The launch of Torpedoboat No. 9 (lately known as H. M. S. *Grasshopper*) at the Chiswick Yard of Messrs. John I. Thornycroft and Co., Ltd., on March 18, is an event of considerable interest from the fact that it is the fifth of the new class of British turbine-driven torpedo boats constructed at this yard, and in addition marks the end of the long and notable list of torpedo craft constructed by this firm at their Chiswick works, commencing with the *Lightning*, known in the Service as Torpedo Boat No. 1, which was delivered in 1877.

The *Lightning* was a boat of 85 feet in length, and capable of a speed of 18 knots, which was then considered most remarkable. Torpedo boat No. 9 is 168 feet long with a beam of 17 feet 6 inches and a draught, under fully loaded conditions, of 5 feet 11 inches. She is fitted with Parsons turbines and Thornycroft boilers using oil fuel, both the turbine machinery and boilers being constructed by Messrs. Thornycroft.

The contract speed is 26 knots, but in the case of the four earlier boats which have already undergone trials, this speed has been exceeded by considerably over a knot. The armament consists of two 12-pounder quick-firing guns and three torpedo tubes. Torpedo Boat No. 8, one of the sister vessels, having successfully carried out all her trials, was handed over to the Admiralty officials at Sheerness by Messrs. Thornycroft on Saturday last. Messrs. Thornycroft now propose to carry

out all work of this nature at their Woolston Works, Southampton, where they are much more favorably situated than at Chiswick.—"Page's Weekly," March 22, 1907.

FRANCE.

The French Naval Disaster.—While the French battleship *Jena* was lying in drydock at Toulon March 12th her powder magazines exploded, completely wrecking the vessel, killing 118 members of her crew, including the commanding officer, Captain Adigar, and Captain Vertier, Chief of Staff of the Mediterranean Squadron, and wounding thirty-eight others, including Admiral Macerou. A child was killed some distance away by an exploding shell from the ship, and many other civilians were injured.

There were three explosions in the interior of the ship, the first occurring at 1.30 P. M., and the others a few minutes later. As yet it has not been learned how the explosion was caused. One statement is that the torpedo men were manipulating a torpedo, which, under some shock, exploded. The violence was so great that two shells in the explosives magazine of 305 millimeters collided and exploded, starting the fire. The gases so produced being compressed in the magazine caused the other explosions. The explosives were hurled immense distances over the town and suburbs. The basin in which the *Jena* was lying to be careened was dry and when the explosion occurred it became imperative to flood the dock to quench the fire aboard and prevent further explosions. The gate was blown in by a shell from one of the warships present.

The *Jena* had just undergone a final inspection of her hull and machinery, the latter having been completely overhauled, preparatory to joining the squadron on March 13th. The crew was in its full strength, being composed of the rear admiral, twenty-four other officers, and 630 men. The magazines had been replenished recently, and contained many tons of both smokeless and black powder as well as a number of charges for torpedoes. The crew of the *Jena* had finished

their mid-day meal only a short time before the explosion, and had dispersed to various parts of the vessel. Most of the men were engaged in work connected with the approaching departure of the ship, but quite a large party had been detailed to attend a lecture that was being given forward.

The French Minister of Marine visited the wreck on March 13th, after the drydock had been emptied of water and the vessel was resting on timber supports. She is not damaged forward, but aft she is terribly injured. There is one especially great rent in the hull. This rent proved a safeguard, as through it the gases caused by the explosion escaped, and thus the vessel was saved from utter destruction. The propellers and helm are undamaged. On board, on the various decks, everything was destroyed. Piles of earthenware mark the dining places, and the débris of woodwork indicates the sites of the cabins. The metal planks resisted the fire, although they are warped and displaced.

Admiral Macerou owes his escape from the fate of other officers, who were burned in their cabins through the blocking of their doors by débris, to the fact that the port holes of his cabin were open when the explosion came.

Various theories as to the cause of the disaster have been advanced. One is that the first explosion was caused by an electrical short circuit, but this is regarded as untenable, and so also is the theory that it was due to the ignition of a shell loaded with black powder. It is thought by many that the accident was due to the spontaneous explosion of smokeless powder. This powder is supposed to be tested every six months, but inquiries among surviving officers fail to establish whether this was done or not on the *Jena*.

Lieutenant Tiercelin, an officer of the *Jena* who was seriously wounded, says in a statement to the Minister of Marine: "I do not believe that the careless handling of anything produced the explosion, for all the men at work were old quartermasters. They were testing the ammunition elevators in the magazine of the 100-millimeter guns. The magazine was under the electric-lighting apparatus, which was below the

officers' quarters, and this fact explains the great number of officers killed. I do not believe that a short circuit was the cause, but solely the age of the powder and its fermentation."

The Toulon correspondent of the "London Daily Mail" quotes a French navy officer as saying that had the accident occurred at sea the vessel must have foundered instantly. Naval experts are astonished by the fierceness with which the flames burst out and raged furiously in the interior immediately after the explosion. That a battleship of modern construction should blaze like a grease box from an explosion arouses the gravest question in reference to naval construction. The correspondent describes the vessel as being as battered and torn as though she had fought a fierce battle. From the forward turret to the stern gallery of the ship is one blistered, riven mass of tortured steel and miscellaneous ruin. Twenty-four hours after the disaster, despite the fact that oceans of water had been pumped between decks, the fire had burned right through, leaving only one-tenth of the entire length of the vessel unharmed. She is almost completely burned inside and out to the very summit of her fore signal-mast. The fire consumed all the boats, ate away the deck planking, destroyed every particle of furniture in the officers' and men's quarters, ruined more than ten guns and seriously injured even the heavier ones. Only the bow guns are undamaged.

Inspector-General Admiral Blenaimé has expressed the opinion that the explosion was caused by the spontaneous combustion of powder, which disintegrates after a certain time and which requires constant inspection for safety. Spontaneous combustion was the cause of the terrible explosion of the *Toulon* powder magazine in 1899.

The *Jena* is a battleship of 12,052 tons displacement, 400 feet in length, sixty-eight foot beams and maximum draught of twenty-seven feet six inches. She was laid down at Brest, January 15, 1898, has three screws, two turrets, two funnels, and has three sets of engines capable of developing 15,500

horsepower and a speed rate of eighteen knots. Her armament consists in part of four 12-inch guns and four torpedo tubes, and her complement number 630.

The destruction of the *Jena* will command the serious attention of naval scientists throughout the world. It is another reminder that modern warships embody elements of construction and the use of materials which are not yet freely understood and which require the most careful study in the interest of safety in peace and of effectiveness in war. The high explosives employed in these great vessels seem capable of undergoing certain chemical changes which make them extremely dangerous even with the most careful handling. The mysterious sinking of Togo's battleship *Mikasa* at Sasebo has been officially attributed to an explosion caused by such changes in her ammunition, and there appears good reason to believe that the destruction of the *Jena* was due to similar causes. Besides these there have been in recent years many other but less serious accidents on large warships in various navies, the causes of which have never been clearly explained—all going to show that the latent properties of the high explosives now in use are not thoroughly known.

Increase in French Submarines.—There are two classes of sub-surface boats in the French Navy, the lines between being well defined; the submarine class is for strictly harbor defense purposes, while the submergible class is rated as seagoing, thus protecting within a greater radius. The two schools have had separate exponents in the French marine, and, except in the cases of *X* and *Z*, there has been no modification from one to the other; these units were submergibles, designed by engineers who had theretofore created submarines, and are regarded as experiments not promising well. There is no example in the opposite direction.

According to the latest information, a total of eighty-nine units have been authorized to be built by the French, of which thirty-one were submarines and the remainder, fifty-eight, submergibles.

The submarines were first in the field with one unit in

1888 and again one in 1893. In 1899 the first "type" appeared, the *Morse,* of which in 1901 two units were finished. In 1901-1902 the ill-fated *Farfadet* type came out with four units. Of the former type one recently sunk, unmanned, and of the latter two units were lost with all on board.

In 1903 the *Perle* type began to appear, and the proposed series of twenty units is not yet complete, it being apparently in question as to completion; this type was based on the *Gustave Zédé* of 1893. Finally there are two units of the small *Guêpe* type building—small reproductions of the *Farfadet* type.

The pioneer designer of sub-surface boats in the French Navy was Gustave Zédé, who designed the submarine *Gymnote;* he was followed by E. Romazotti with a submarine named for his predecessor, the *Gustave Zédé;* later came his *Morse.* Engineer Maugas designed the *Farfadet* type, and lastly M. Petithomme the *Guêpe* type. As the pioneer boat is out of service, and the little *Guêpes* are not yet finished, it may be said that the entire submarine class in the French Navy today is built on the designs of Romazotti and Maugas. It may be noted that each of these designers produced one submergible experiment, neither of which has been duplicated up to the present time, they being the *X* and *Z* mentioned earlier in this summary. Though, and perhaps because of, coming later, the submergible class is much more numerous in the French Navy than the submarine class. In 1899 Engineer Laubeuf produced the *Narval,* which has been the prototype of the entire class of French submergibles; in 1901 four more units of this type were launched.

In 1904 the *Aigrette* type appeared—two units—and there are now building in the various French naval stations twenty-one of this design, with three more series, aggregating twenty units, projected and included in the totals cited above. In 1905 Bertin designed an experimental submergible along Laubeuf lines, and also the *Omega,* which is styled an "official development" of the Laubeuf type; on this *Omega* type there are now being constructed six units, known as the *Emeraude*

type, making a total of Laubeuf submergibles of fifty-six out of a total in the class of fifty-eight, the other two being the Romazotti and Maugas experiments.

It may be justly assumed that the submergible class is preferred in France from the fact that of eighty-nine existing and projected sub-surface units fifty-eight are submergibles and thirty-one submarines—an even greater disparity promises, from recent developments, to obtain in the future; so far no disaster has ever occurred in one of the submergible class. The development of sub-surface boats in France has outstripped numerically that of any other country many times over; in the United States, for instance, we have in all twelve units, including those at present under order—less than one-seventh of the number of the French, with many times the number of harbors to defend. The curve of displacement, chronologically considered, has been extremely irregular in the submarine class, while in the submergible class there has been a steady increase in tonnage, as will be seen from the appended condensed table:

Submarines.	Tons.	Submersibles.	Tons.
1888, *Gymnote*, . .	30	1899, *Narval*, . . .	200
1893, *Gustave Zédé*, .	270	1904, *Aigrette*, . .	250
1899, *Morse*, . . .	146	1905, *Omega*, . . .	375
1901, *Farfadet*, . .	185	Bldg., *Emeraude*, . .	430
1903, *Perle*, . . .	68	"*Q*," 53–60, .	500
Bldg. *Guêpe*, . . .	44	Projected, ...	800

Since the data was collected on which the above statement is based Chief Engineer Laubeuf has resigned from the French Navy, and the present Minister is now proceeding to build several units of largely increased tonnage in the submergible class.

ITALY.

The Italian Navy.—The Italian naval authorities have decided to sell or break up a number of battleships and other vessels during the five years 1907-12. The list comprises 21 ships of various classes, including the *Duillio* (launched in 1877) and the *Andrea Doria* (launched in 1891), as well as

51 torpedo boats. With the proceeds, which are estimated at £250,000, the Italian Government intends to make large purchases of coal for its navy.

Submersible Boat "Glauco."—The *Glauco* is the first of a series of five submersible torpedo boats which the Italian Navy will have completed in the course of a few months. Their displacement when fully emerged is 175 tons, and 220 tons when completely submerged. They are designed for a speed afloat of 15 knots, using the explosion motors only, and for a cruising speed of 10 knots. The fuel carried is sufficient for a run of 175 miles at the higher speed, and for 600 miles at the 10-knot speed. The time required for diving when fully emerged is five minutes. Each boat carries two torpedo tubes for 18-inch torpedoes. Their length and breadth at waterline afloat are, respectively, 42 meters and 4.3 meters (137 feet 9 inches and 14 feet 11 inches. The reserve of buoyancy when fully emerged is said to be 120 tons.

Engineer C. Laurenti is now Technical Director of the Fiat-Muggiano Company, Spezia, which, we are informed, has in hand at the present time several submarine boats embodying improvements on the *Glauco* type.

ROUMANIA.

Torpedo Vedette Boats.—Eight screw torpedo vedette boats have recently been built for the Royal Roumanian Government by the Thomas Iron Works, Shipbuilding and Engineering Company, London. The vessels are 100 feet long, 13 feet beam, with a draught of water of 2 feet 9½ inches, and a displacement of 51 tons.

The hulls are built throughout of steel, the deck and sides to just below the water line being bullet-proof. The vessels are each fitted with a 47-millimeter gun on top of the conning tower forward, and a small mitrailleuse gun aft. Gear for dropping torpedoes over the side is fitted on both sides amidships, and two spar torpedoes are fitted forward over the bows. The boats are also supplied with powerful search-

light projectors, and are lighted throughout by electricity. Accommodation is provided for four officers and twelve men.

The problem set the Thames Company was an extremely difficult one, the maximum length allowed being 100 feet; maximum draught, 2 feet 9½ inches, and the speed 18 knots on a trial of four hours duration, under penalties. Owing to the abundant supply of oil in Roumania it was also desired to use petrol residuum as fuel. It was found impossible to use internal-combustion engines on account of their excessive weight, so it became necessary to provide the 600 indicated horsepower on a weight of 16 tons by using steam propelling machinery.

There are two sets of compound engines of the inverted type, the twin screws running in tunnels formed in the stern of the boat. The cylinders are 8½ inches, and 17 inches in diameter respectively; with a stroke, 9 inches, and they are supported on turned-steel columns strongly braced together. The main bearing frames are separate castings of Stone's high-tension bronze of 30 tons per square inch tensile strength, bolted to long steel girders, which also carry the thrust block, feed pumps and air pumps. The bearings throughout are lined with white metal, and centrifugal lubricators are fitted to all the revolving parts. The crank shafts are solid, and balance weights are fitted to the cranks.

There are two feed pumps fitted to each main engine, these being worked from a countershaft driven by worm and worm wheel from the crank shaft at a reduced speed, the pumps making one revolution to four of the engines. Each pump has a separate connection to the boiler through two non-return valves, and hand-controlled valves are fitted on the suction side only. This method of working is very simple, and is found to render the feeding of the boiler very safe and easy. There is also no danger of burst feed pipes owing to feed valves being inadvertently closed. The air pumps are driven off the forward end of the crank shaft through a universal joint, and run at the same speed as the main engine. Each set of machinery is complete with its own condenser, and the centrifugal circu-

lating pump, steam, exhaust, feed and other pipes are all in duplicate, so that each main engine is entirely independent of the other.

The circulating pump has an impeller 12 inches in diameter, driven by a single-cylinder engine. The pumps are entirely of gun metal, and from each pump there is driven an auxiliary air pump, which is so fitted that it can be stopped whilst the main air pump is working. The circulating water passes once through the condensers, and the condensers and pipes are arranged so that, with the circulating pumps stopped, enough water will pass through the condensers to maintain a vacuum of 22 inches, or 23 inches. There is a hot-well tank at the forward end of the engine room, and a reserve feed tank at the after end, both being provided with filters. An auxiliary feed pump, of the direct-acting type, by Messrs. Caird & Rayner, is fitted in the engine room. It has a cylinder 7½ inches in diameter, and a pump 5½ inches in diameter, with a stroke of 8 inches. This is large enough to feed the boiler at full power, and will do duty for bilge or deck, and will pump through the filter on the reserve tank. The propellers are of bronze, 3 feet 3 inches in diameter, three-bladed, accurately pitched and balanced. Steam is supplied by a Thames Iron Works water-tube boiler. The steam pressure is 185 pounds per square inch, and the heating surface 1,290 square feet. The furnace is specially arranged for burning oil fuel, the ordinary fire bars being replaced by fire brick, with slots for the admission of air. The casing is carried down under the hearth, and the air is admitted directly at the front, and at the back through a casing which serves the double purpose of keeping the back cool and of heating up the air before entering the furnace.

Owing to the very limited weight available for the machinery, it was necessary to resort to steam spraying. Four special burners are fitted to each boiler, two at the top having an angle downwards, and two on the bottom, at each side of the fire door, directed towards the center.

Oilbunkers are fitted on each side of the boiler, and a steam pump is fitted for pumping from them to the present-use tank.

A steam coil is fitted in each tank for heating the oil, the steam running first through the present-use tank, and then through the coils in the bunkers. The water from these coils is run into the bilge to obviate any danger of getting oil into the feed water. Both bunkers and the present-use tank act as settling tanks, and a valve is fitted at the bottom of each to drain off water as it collects. There is not enough height in the boat to run the oil through the burners by gravity; the present-use tank is therefore worked under an air pressure of from 1 pound to 10 pounds, the air being supplied by a small pump driven off the fan engine, and a reducing valve regulates the pressure.

The forced-draught fan is 4 feet in diameter, driven by a single steam engine running at 800 to 1,000 revolutions per minute, the air pressure at full power being from 4 inches to 5 inches of water. For the purpose of securing good ventilation in the engine room the air inlet of the fan is through the upper part of the engine-room bulkhead, and the mushroom-topped ventilator at the aft end of the engine room. The air for the boilers has thus to pass right through the engine room, serving admirably to ventilate that compartment. In the steering compartment is the generating set for lighting the ship and working the projector. The 8-kilowatt dynamo was made by the Thames Iron Works electrical department, and the enclosed tandem compound engine, made by Messrs. Peter Brotherhood, Belvedere road, Lambeth, S. E., has forced lubrication.

Four bilge ejectors, each of 8 tons per hour capacity, are fitted, one in each compartment. Chadburn's reply telegraphs are fitted from conning tower to both engines.

The boats were designed by the late Mr. G. C. Mackrow, and the official trial of the first boat took place on the Thames, September 25, 1906. The Roumanian Government were represented by Major Demetriade, Director of Marine; Captain-Commander Balesco, President of the Commission; Captain Negru, Captain Mihail and Captain Stefanesso.

The trial was of four hours' duration, and the following results were obtained:

Mean speed, knots	18.0365
indicated horsepower	622.7
revolutions per minute	554.8
Boiler steam pressure, pounds per square inch	182.3
Air pressure in stokehold, inches water	4½
Pressure of steam in oil-fuel burners, pounds per square inch	181
oil for oil-fuel burners, pounds per square inch	7
Temperature of oil in warming tank, degrees Fahrenheit	175
Oil consumption, pounds per I.H.P. per hour	1.73

This result was obtained with the two top burners only in operation. The power was maintained steadily, and generally there was absolutely no smoke or visible gases from the funnels.

The machinery, which was designed by Mr. R. Warriner, ran smoothly and without vibration; no sign of heating was observed in any of the bearings, although only the ordinary oil service was used. On the stopping and starting trial the engines were easily and rapidly manipulated by one man. Four of these boats left the Thames Iron Works last October, taking altogether six weeks on their journey from London to Galatz through the waterways of Central Europe. They were not allowed to proceed under their own steam, but had to be towed along with the barges, etc., in the usual way. The last four boats left the works in March, and are now on their way through the center of Europe.

PERU.

New Peruvian Warships.—The *Coronel Bolognesi,* with her sister ship *Almirante Grau,* are the pioneers of the modern Navy which the Peruvians desire, and the latter ship had just returned from her trials when the *Coronel Bolognesi* was launched. These two cruisers are somewhat similar in design to the British scout class, although their armament will not enable them to stand up against heavily armored vessels. In combination with high speed, it is to enable them to harass heavy fighting ships, and at the same time to meet attack by torpedo boats.

The trials of the *Almirante Grau,* which have been con-

cluded, were arranged to insure that the vessel should be able to steam at sea under all weather conditions at twenty-two knots, to make when necessity arises twenty-four, and to be able to cruise at a low speed with a light consumption of fuel. On the trials on the Clyde the vessel averaged, on full power, over twenty-four knots with the boilers working under normal conditions; the coal consumption was lower than the average of the eight new scouts for the British Navy; and no difficulty was found in realizing twenty-two knots with only eight of the ten boilers.

The trials occupied a week on the Clyde. The first test was a series of progressive speed trials over the measured mile at Skelmorlie, and the results were up to expectations. On the measured mile she steamed 24.64 knots, and considerably exceeded twenty-four knots throughout the full-power trial, the mean power being 14,144, while the revolutions were 216 per minute. The Admiralty "constant" was 229. The continuous sea-speed trial was equally satisfactory. The Peruvian Government desired that this trial of twenty-four hours' duration should be divided into four successive periods of six hours at progressive powers, viz: 900 horsepower for the first six hours, 3,600 for the second six hours, 6,000 for the third six hours, and 8,500 for the fourth. The coal consumption at low powers is, of course, higher, because the proportion of steam to be generated for the auxiliary machinery is greater than on a high-power trial. At low power more than one-third is needed for auxiliaries, whereas on a full-power run the proportion may be less than one-tenth. The coal consumption ranged between 2.27 pounds per horsepower for the low power to 1.635 pounds for the high power—all within the guaranteed conditions. This coal-consumption trial was completed at Barrow.

The *Coronel Bolognesi,* on trial, made 24.726 knots, with the engines indicating 14,384 horsepower, at 218.7 revolutions per minute. The Admiralty "constant" was 228.

Each ship has a bow and stern-chasing gun of 6-inch caliber, with an arc of train of 270 degrees, and in addition there are

eight 14-pounder quick-firing guns, eight 1¼-pounder guns, with two submerged tubes for firing 18-inch torpedoes. She has for protection a deck of 1½-inch thickness extending the full length, and the conning tower is of 3-inch armor. The armor and armament have been made at the Vickers Works, at Sheffield.

The engines are of the twin-screw four-cylinder triple-expansion type, balanced on the Yarrow-Schlick-Tweedy system, and designed to develop 14,000 horsepower with steam at a working pressure of 250 pounds at the engines and 280 pounds at the boilers. There are ten water-tube boilers of the small-tube type, arranged in three separate watertight compartments, to work under forced draft on the closed-stokehold system.

Special care has been devoted to the natural and artificial ventilation of the ship, and to the heating, steam radiators being extensively employed. There is, of course, a complete installation of electric light. Although the ship is small, baths are provided in all the living quarters. Refrigerating plant and ice-making machinery, with cold-storage chambers adjoining, are notable features. There is also a mechanically-equipped bakery on board, so that the vessel excels many warship of much greater size in the elements conducing to comfort.

The *Almirante Grau* will be the flagship of the embryo Navy, until a more powerful ship is provided. Both cruisers are named after Peruvian heroes. The principal difference in the design of the two is that the *Almirante Grau* has a full poop to provide quarters for the admiral; in the *Coronel Bolognesi* there is only half a poop. Both ships have a length of 370 feet and a breadth of 40 feet 6 inches. The displacement of the *Coronel Bolognesi* is about 3,180 tons, with a draught of 14 feet 2 inches, and a block coefficient of 0.525; the *Grau* draws an inch more, and displaces 3,200 tons. The design called for 24 knots, with 14,000 horsepower.

MERCHANT SHIPS.

Japanese Cable-Steamer "Ogasawara Maru."—The cable-steamer *Ogasawara Maru,* which left Nagasaki on October 5th last for the cable-laying work between Tsushima and Kiushiu is the first ship of its class built in Japan. This steamer is owned by the Department of Communications of the Japanese Government, and was constructed at the Mitsubishi Dockyard and Engine Works, at Nagasaki. The design of the vessel was entrusted to Dr. C. Shiba and Mr. K. Suyehiro, professors of the Engineering College of the Imperial University of Tokio; the contract was placed June 1, 1905, and the work commenced in November of the same year; the vessel being launched on June 2, 1906, six months after the laying of the keel. She is a steel spar-decked twin-screw steamer of 1,455 tons gross, with engines developing 1,850 indicated horsepower at full speed. The hull and machinery were constructed under the Japanese Government's Shipbuilding Encouragement Law. The principal dimensions of the vessel are:

Length between perpendiculars, feet and inches	240-0
Breadth, molded, feet and inches	34-0
Depth, feet and inches	22-3

The guaranteed speed was 12 knots, the ordinary sea speed expected being 11 knots. During the official speed trials the maximum speed obtained was 13,286 knots, and the average of six runs between the two measured-mile posts, three nautical miles apart, was 13.085 knots.

The hull and machinery of the vessel were all constructed by the Mitsubishi Company, except the cable gear, which was supplied by Messrs Johnson & Phillips, of Charlton, Kent. Three cable tanks are provided in the vessel: the fore tank, of 20 feet 6 inches diameter, intended for the storage of shore-

end cables; main tank, 27 feet in diameter; and after tank, of 23 feet diameter. The total capacity of the three tanks gives storage for 600 tons in all of deep-sea cables. Each tank is provided with a cone at the center for storing battens or fresh water. The hatchways of the tanks are provided with girders, carrying bell-mouth crinolines, etc.

The structural arrangements provide two complete decks running fore and aft, and an inner bottom carried nearly the whole length of the vessel, a feature of the structure necessary in this class of vessel. The hull is subdivided into five water-tight compartments by transverse bulkheads.

The foremost cable tank, provided for the shore-end cables, is carried up to the level of the main deck. The main tank is brought 5 feet above the main deck, leaving ample clearance between the top edge of the tank and lower side of the deck above. The bottoms of these tanks rest on the top of the inner bottom plating. The after tank is similarly constructed to the main tank, the bottom, however, resting on the top of the shaft tunnel.

The vessel is rigged as a two-masted schooner. Besides the necessary cargo appliances, the foremast is provided with yards to facilitate the lowering of buoys, etc., in cable-laying work.

In front of the foremost cable tank, and on the main deck, the cable gear is located, as shown on the plan; this part of the deck is most carefully constructed for minimizing the vibrations caused by the motion of the cable engines.

The double combined picking-up and paying-out machine, which is placed forward of the foremost tank, is very similar to that fitted in the cable steamer *Pacific,* and was supplied by Messrs. Johnson & Phillips. The machine is fixed on the main deck, with the winding drums appearing through an opening in the spar deck, all the handles for controlling the machine being brought to the spar deck. This double machine practically comprises two complete and independent gears, both of which may be used either for picking up or paying out. This double gear is provided with two sets of steam engines, placed fore and aft of the gear; either of

the engines can drive the gear for picking up and also for paying out when the pull on the cable is moderate. When a heavy load is on the cable both engines are put into work conjointly.

The drums can be driven at three different speeds by changing the gear wheels by means of clutches, which action can be performed from the starting platform on the spar deck. In the ordinary cable-laying or picking-up jobs, one of the engines may be kept as spare, since the power of one only is always quite sufficient for ordinary working. This reduces to a minimum the risk of total disablement of the gear. The cable-drums are 6 feet in diameter by 21 inches between the flanges, and they have steel gearing teeth and brake rings bolted to the rims of the drums for driving and holding, the drums being arranged to run loose on the shaft. The main brakes encircling the rings attached to the drum consist of steel bands, with elm-block liners. They are made to close by means of right and left-handed screws actuated through worm gearing from handle wheels mounted on the stands on the starting platform. This arrangement gives very fine adjustment for the lighter load on the cable, and at the same time provides enormous holding power when necessary. To the back of each brake strap is fitted a water-service pipe, so as to cool the rubbing surface of the elm-blocks and drum. This cooling water is supplied from an independent small steam pump, which is arranged to be started or stopped from the starting platform. Both brakes can be coupled to each drum when an excessive load is on, and give a very powerful holding-back action to the machine.

The hauling-off and holding-back gears stand on the spar deck, and are driven from the geared drum by means of endless chain through a friction coupling and free wheel. These gears are made to traverse across the face of the drum by means of screws, in order to lead the cable to any position on the drum. There are three sheaves in the bow gear. They have **U** section, run loose on a fixed shaft, cast-iron whiskers or guards being provided between the various sheaves

and at the outsides to prevent the cable from getting out of the grooves. The stern gear has only one cast-iron sheave mounted on a girder fixed on the spar deck, and is also provided with whiskers. For registering the pull on the cable three dynamometers are provided, these being fitted with Messrs. Johnson & Phillips' patent arrangement of internal spring, and their maximum readings are 25 tons for picking-up purposes, and 5 tons for paying out. The usual cable leads of **V**-shaped sheaves, mounted on **A** frames, are provided fore-and-aft to the dynamometers.

The cable engines were tried under steam outside Nagasaki Harbor. The machine wound up the load of 25 tons, consisting of bundles of anchor cables, from the bottom of water 17 fathoms deep, and at the speed of one turn per 22 seconds, which is equivalent to 1.02 knots lifting speed of cable. In this case of heavy pull the two sets of engines were used conjointly. For the purpose of turning over the cable from one tank to another, or taking in cable from outside, two sets of electric hauling gears, driven by a 6-brake-horsepower motor of closed-in type, were supplied by Messrs. Johnson & Phillips; they are driven off the electric main on the spar deck.

The propelling machinery of this vessel consists of two sets of triple-expansion engines, constructed under the Japanese Government's Shipbuilding Regulations, 14¾ inches, 24 inches, and 40 inches in diameter, with a 27-inch stroke. The propellers are 11 feet in diameter, and have each three blades of manganese-bronze.

The steam-generating plant of this vessel consists of two single-ended Scotch boilers and one donkey boiler of the Cochran type. The main boilers are 14 feet in diameter and 11 feet 6 inches in length, with three furnaces of 3 feet 6 inches internal diameter for each. The flues are of Deighton's patent corrugated pattern, united to the combustion chamber with detachable ends. The working pressure of the main boilers is 185 pounds, and that of the donkey boiler 100 pounds per square inch.

One evaporator of 10 tons capacity, supplied by Messrs.

Caird & Rayner, of London, is fitted, with its accessories; it is so arranged that the steam produced may be admitted to the main and auxiliary condensers, and also to the intermediate-pressure receiver of the main engines.

The general-service and ballast pumps are of the usual Worthington type, and all of them are so connected that each is at any time able to send water to any of the cable tanks.

The electric installation of this ship consists of two sets of steam dynamos, supplied by Messrs. Clarke, Chapman & Co., of Newcastle-on-Tyne, working at 110 volts. A searchlight projector of 16,000 candlepower is fitted on the flying bridge. A refrigerating machine of the carbonic-anhydride system, supplied by J. & E. Hall, of England, is fitted in the port wing of the engine room, on the same seat as the steam dynamos. This machine is for cold storage and for manufacturing ice. A Kelvin sounding machine is provided, a James submarine sentry, and also a steam-driven sounding machine for lifting samples from the ocean-bed, as usual in cable steamers.

In boat equipment the steamer is provided with two sampans, two 22-foot steel life boats, and one steam cutter, 30 feet long. The steam cutter is of the Navy type, fitted with a water-tube boiler and one set of compound engines, capable of developing sufficient power to propel the boat at 7 knots.

As will be seen from the above description, the cable steamer *Ogasawara Maru* is constructed and equipped in a very efficient manner; she was officially tried under steam on July 30th last, on which occasion she went through her various tests to the complete satisfaction of the owners' representatives.—"Engineering."

Wreck of American Steamship "Dakota."—The great 20,714-ton American steamship *Dakota,* sister ship to the *Minnesota,* the largest ever built in America and among the largest in the world, struck on the reefs off the coast of Japan, 40 miles south of Yokohama, on March 2, 1907, and now lies partially submerged, one mile from shore. The passengers and crew were landed in safety, but their belongings were lost. The Japanese steamer *Omimaru,* which went to the assistance

of the disabled liner, returned to Yokohama, and reported being unable to get alongside the wreck owing to the heavy seas which were running.

The *Dakota* and her sister ship, the *Minnesota*, belong to the Great Northern S. S. Co., and operate on the line between Seattle and the Orient, calling at Yokohama, Japan, and Hong Kong, China, and occasionally making trips to Manila.

They were constructed by the Eastern Shipbuilding Co., at Groton, Conn., during the years 1903-4-5, the yard being erected expressly to build them. When they went into commission they were the largest—although not the longest—vessels in the world, a distinction which has since passed to the newer of the British-built trans-Atlantic lines. The *Dakota* is 622 feet between perpendiculars, 650 feet over all, 73½ feet beam, and 19 feet depth of hold, with a gross tonnage of 20,714 and net tonnage of 13,305. She is driven by twin screws, carried a crew of 253 and had accommodations for several thousand passengers, and an immense quantity of freight.

The *Dakota* cost $3,000,000 and is insured for $2,500,000.

Wreck of the "Berlin."—Early on the morning of February 21st, the twin screw steamship *Berlin*, belonging to the Great Eastern Railway Company, was wrecked at the mouth of the waterway leading to the landing station at the Hook of Holland. The *Berlin* was on her way from Harwich, and her voyage was nearly over when the disaster occurred. A full gale was blowing at the time and the waves were mountainous. The entrance to the waterways is between two jetties running out into the sea. These jetties are nearly parallel, and their line is from a little north of west to a little south of east. The navigable channel between them is not quite central, and though the actual water surface is some 2,250 feet wide, the deep water channel in which there is an average of nearly 30 feet, is in places not more than 300 feet in width. Elsewhere the water shoals, and in places is not more than 8 feet or 9 feet deep. The deep channel is nearer the north jetty than the south, and its direction is indicated to the navi-

gator during the day by means of buoys, while at night it may be followed by keeping two lights in line. The direction of the wind was such that the seas would have been beating right in between the two dams and against that on the north side. From some cause which has not been at present discovered, the ill-fated vessel on making for the channel struck the end of the north breakwater or dam and then turned broadside to the sea. Very soon she broke in half and the fore portion foundered, carrying with it numbers of the crew and passengers. The stern portion remained on the sand, and was continuously washed by enormous waves. Heroic efforts were made by the Dutch lifeboatmen and others, the Prince Consort himself taking an active part in the relief operations, but in spite of everything that could be done, only fifteen were brought to shore alive, and these in a pitiful plight. It is feared that nearly 130 persons lost their lives in this terrible accident—one of the worst which has visited Europe for many years.

OBITUARY.

CHARLES HAYNES HASWELL.

Charles Haynes Haswell, the oldest civil engineer of note in the world and one of the most distinguished marine engineering experts, died at his home in New York City, May 12, in his ninety-eighth year, his death being due to a shock attending a dislocation of the shoulder, which he had sustained the day before in slipping and falling on the floor of his house in West Seventy-eighth street. Up to the time of his accident he was in good health, and, although so nearly one hundred years of age, he was regularly at his work in charge of important construction operations for New York City.

Mr. Haswell was born of English parents in New York, May 22, 1809. At the age of 19 he finished his schooling, which had been along classical lines, and, having inclinations toward a technical career, he secured employment in the engine works of the pioneer establishment of James P. Allaire. While with this firm he was selected by the United States Navy Commissioners to take charge of the building of a steam war vessel for the Government and in 1836 joined the United States Navy as chief engineer, receiving his commission as engineer-in-chief in 1845. Some years prior to this date he conceived the idea of building a steam launch and in 1837 constructed the *Sweetheart,* which was the first private steam yacht launched. He held the rank of engineer-in-chief of the Navy until 1851, and during that period designed the machinery for ten war vessels and introduced numerous mechanical improvements for increasing the early steam navy's efficiency.

In 1847-48 he designed the entire boiler and engine equipment of the steam frigate *Powhatan.* Owing to a lack of professional aid and the urgent requirements of the service and

the contractors, he was compelled to personally design every detail and made the working drawings himself in the intervals between attention to the necessary duties of his office as engineer-in-chief. The design of the engines was novel in several respects and wholly so in the fact that they were set in wrought-iron frames, the first construction of its kind. The *Powhatan* had a length of 250 feet and a displacement of 3,600 tons. The engines were inclined and the two cylinders were 70 inches in diameter, the stroke being 10 feet. The engines were designed to run at 13.5 revolutions per minute. The steam pressure was 10 pounds per square inch, the indicated horsepower 1,100. The paddle wheels of this man-of-war of 60 years ago were 31 feet in diameter and the speed of the ship about 10 knots.

After retiring from the Navy in 1851 Mr. Haswell built several merchant steamships, and then engaged in engineering practice in New York. For over 40 years he was surveyor of steamships for the marine underwriters of New York. He designed and located the buildings on Hoffman Island in the lower bay and built the crib bulkhead at Hart's Island. He constructed what at the time were the most extensive and difficult building foundations in the city. For several years past he had supervised extensive work of construction and improvement at Riker's Island. From 1855 to 1858 he was a member of the board of councilmen of New York City, and during the latter year was its president. Later he entered the employ of the city, and during his many years of service acted as chief engineer of the dock department, superintending engineer to the department of charities and correction and the board of health, and consulting engineer to the board of public improvement. He was appointed by the latter body in 1898 to design and supervise the extension of Riker's Island, and in 1902 was appointed consulting engineer to the board of estimate and apportionment, which position he held at the time of his death.

Mr. Haswell first brought out his "pocket-book," which is now in its seventy-second edition, in 1843, the volume at that

time containing 284 pages. He was also the author of "Reminiscences of an Octogenarian," a book of memoirs covering the old New York from 1816 to 1860. He was a member of the Engineers' Club of New York and Philadelphia, and belonged to many engineering and scientific organizations, among these being the American Society of Mechanical Engineers, the American Society of Civil Engineers, American Society of Naval Engineers, the Institution of Civil Engineers and the Institution of Naval Architects of Great Britain.

WILLIAM DURELL STIVERS.

William Durell Stivers was born in Jersey City, N. J., February 20, 1871, received his education in the public schools, graduated from the High School in 1887, and pursued some special studies in mechanical engineering at the Cooper Institute in New York. He entered the DeLamater Iron Works in 1887, and was assigned to special service in the superintendent's office where he had unusual opportunity for acquiring a special training in shop management, engineering and experimental work. His great love for the profession—probably imparted to him in part by his uncle, the illustrious Chief Engineer B. F. Isherwood, U. S. N., with whom he was a great favorite and who took deep interest in his development—and his indefatigable diligence and interest in all mechanical pursuits caused him to make rapid strides, and he soon became useful in many ways during the various interesting experiments that were at all times conducted at the DeLamater Iron Works, as, for instance, those of the noted Ericsson expansion engine, the hot-air engine, the Belleville boiler, refrigerating and compressed air machinery and other constructions. After the dissolution of the DeLamater Iron Works in 1889, William Stivers entered the Quintard Iron Works as draughtsman, eventually rising to the position of Acting Superintendent. He supervised the building and installation of the machinery of the U. S. S. *Maine*, which was afterwards destroyed in Havana Harbor. He was the Works' representa-

tive on the trial trips of the U. S. S. *Concord, Bennington, Detroit* and *Marblehead.* He also had a prominent part in the trial trip of the U. S. S. *Bancroft.* In 1902 he left the Quintard Iron Works to accept the position of Superintendent of the Yonkers Works of the Otis Elevator Company, which position he held until March, 1904, when he was engaged by the C. W. Hunt Co. as Executive Engineer, remaining active in that place until his death, December, 1906. He was married in 1898 and leaves a widow.

He joined the American Society of Naval Engineers in 1904. A. H. R.

CAPTAIN GEORGE H. KEARNEY, U. S. N.

By Commander John R. Edwards, U. S. N.

The death of this sterling officer was a distinct loss to the Navy. From the time he entered the Service, he commanded the respect of his seniors, the friendship of his messmates, and the obedience and esteem of his subordinates. During his whole career in the Navy he was regarded as an exceptionally strong man, whether viewed from the standpoint of engineering ability, general knowledge, mental attainments or unimpeachable moral character.

The fact that he was invariably assigned to important duty best tells of his professional esteem and ability. His character was such that there was ever manifest in his performance of duty those attributes and qualities which made him a strong example of the best traditions and spirit of the Service. His high sense of honor and splendid manhood fittingly supplemented his capacity of leadership. His reserve, combined with unselfishness of purpose, likewise prompted him to avoid rather than to seek honors that were his due, but in all matters pertaining to naval engineering policy and progress his judgment and experience were sought and followed.

But few officers possessed more moral courage than he, although he never placed himself in an attitude where it was necessary to exhibit such attribute until after earnest and mature consideration had been given to a subject.

His professional life-work is best reflected in the efficient and splendid manner in which engineering duty, both afloat and ashore, is now being performed by officers who were cadets at the Navy Academy when Captain Kearny was head of the Department of Marine Engineering and Naval Architecture at that institution. His character, integrity and manhood have been stamped upon those whose privilege it was to serve under him. The full measure of his worth as an officer and a man was best known to those to whom he was not reserved and to whom he could freely speak of his life's work and purpose.

BOOK REVIEWS.

INTERNAL COMBUSTION ENGINES. An Elementary Treatise on Gas, Gasoline, and Oil Engines, for the Instruction of Midshipmen. By JOHN K. BARTON, Commander, U. S. Navy. Size, 9⅛x5¾ inches; pages, 135; illustrations, 52; price, $1.50. Annapolis, Md., 1907, THE UNITED STATES NAVAL INSTITUTE.

Commander Barton has made a very timely and appropriate addition to the series of modern American text-books for use in the engineering department of the United States Naval Academy.

This book, however, is not to have its usefulness confined to the class rooms and experimental laboratories at Annapolis. It appears at a time when the field of application of internal-combustion motors is being rapidly and satisfactorily extended to marine purposes, chiefly as a result of the employment of hydrocarbon fuels which are removing many of the limitations hitherto imposed afloat.

In our own and in foreign services, extensive experimental research is underway and higher powers together with simpler and more compact installations are positively assured. Already many torpedo boats, submarines and tenders are operating with these motors.

The work at hand reviews the principal types of gas and oil engines in construction and operation.

The chapter on Gaseous Fuels and Producers is especially important since the marine producer is attracting most attention in the application of the motor to naval uses. The chapter on Gas Engine Tests renders the work thoroughly useful in the laboratory and is comprehensively arranged. It comprises the preparation of the data for and the procedure in an engine test, and includes, in tabular form, the report of such test and

also the report of a heat-balance test. The arrangement of apparatus for the determination of the calorific value of the fuel by means of the Junker Calorimeter is included.

The final chapter deals with the care, management and practical operation of the motors, with a mention of the troubles to look out for and how to remedy or avoid them.

The illustrations are good and the text clear—a boon to students.—W. T. C.

DIE SCHIFFSSCHRAUBE (The Marine Screw Propeller). By ALBERT ACHENBACH. Part II. Kiel. ROBERT CORDES.

Part I of this work, which is to comprise three parts, was noticed on page 360 of Volume XVIII. Part II, covering, with the appendix, 152 pages, with 20 plates, is devoted almost entirely to the construction of screws for both naval and merchant vessels. There are chapters devoted to the drafting, to material, and to the work in the foundry and machine shop, with tables of the dimensions of screws actually built for different classes of vessels; a separate chapter is given up to the Niki propeller. Although there is little that is new in this Part, the material is well presented and is comprehensive.

The appendix, of 40 pages and 5 plates, covers the screw arrangement in motor boats, and includes descriptions of couplings, feathering screws and reversing gear.

One disadvantage of publishing this work in Parts is apparent in Part II, where a few references are made to plates embraced in Part I.—R. S. G.

THE STEAM ENGINE AND OTHER HEAT MOTORS. By W. H. P. CREIGHTON, U. S. Navy (Retired), Professor of Mechanical Engineering, Tulane University of Louisiana. JOHN WILEY & SONS, New York.

This excellent work on the Steam Engine and Other Heat Motors, containing 500 pages, written particularly as a textbook, will be found of very great value to the practical engineer. Mr. Creighton, from his long experience, both as an

Engineer Officer in the Navy and as an instructor in mechanical engineering, is especially equipped for producing such a work as we have before us. The book is divided into eighteen chapters, with an appendix, as follows:

Chapter I.—Review of Elementary Principles and General View of Steam-Engine Plant.
II.—Steam-Engine Indicator and its Calibration.
III.—Curves and the Work of Expansion.
IV.—Zeuner and Bilgram Valve-Diagrams and Design of Plain Slide Valves.
V.—Measuring the Effects of Heat.
VI.—Measuring the Effects of Water and Heat on Water and Steam.
VII.—Measurement of Heat Losses.
VIII.—Entropy.
IX.—Condensers and Air Pumps.
X.—Small Auxiliaries.
XI.—Multiple-Expansion Engines.
XII.—Revolution Control.
XIII.—Speed-Variation Control.
XIV.—Steam-Engine Tests.
XV.—Superheated Steam and Steam Turbines.
XVI.—Gas Engines and Gas Producers.
XVII.—Boiling in a Vacuum.
XVIII.—Refrigeration.

Appendix.

Table I.—Properties of Familiar Substance.
II.—Hyperbolic or Naperion Logarithms.
III.—Heating Values of Various Substances.
IV.—Oxygen and Air Required Theoretically for the Combustion of Various Substances.
V.—Relative Humidity. Per cent.
VI.—Weights of Air, Vapor of Water, and Saturated Mixtures of Air and Vapor at Various Tempertures and Constant Pressure.

Table VII.—Entropy of Water and Steam.
VIII.—Saturated Steam.
IX.—Mean Pressures for Various Methods of Expansion.
X.—Mean Pressures for Various Methods of Expansion.
XI.—Mean Pressure Ratios.
XII.—Terminal Pressure Ratios.
Entropy Diagram.
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ERRATA.

U. S. S. "KANSAS"—*May*, Vol. XIX, No. 2.

Top of page 432, Revolutions port engine, for 130.52 read 120.52.
I.H.P. main engines, for 1,930 read 19,302.

Page 435, 12th line, Speed of ship in knots per hour, for 18.004 read 18.094.

Page 462, 10th line, G. S., for 1,000 read 1,100.